RIPPER'S HEAT ENGINES

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NEW EDITION

Revised by

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PREFACE TO THE EDITION OF 1939

THE continued popularity of this text-book is evidence that the general scheme is appreciated and that the combination of clear explanations of essential principles and an outline of their main applications satisfies the needs of a large number of students.

Rapid and extensive changes in practice have occurred in recent years. The steam turbine on the one hand, and the internal combustion engine on the other, have displaced the reciprocating steam engine from the position it held as the principal prime mover both in land and marine practice ; and the practice in application of such details as governors, valve gears and reversing gears for steam engines has altered, the older types having become obsolete or obsolescent.

A number of revisions and additions, with omissions of obsolete matter, have been made in this edition in order to bring the book up to date so far as its scope as an introductory text-book extends, and a new set of examples will, it is hoped, further enhance its usefulness.

Acknowledgment of sources of additional information are made in the text, and thanks are due to firms and persons who have supplied this information, also to The Institution of Mechanical Engineers for permission to reproduce Figs. 189 and 190.

A. T. J. K.

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CHAPTER I

HEAT AND WORK

A heat-engine is a means of converting heat into useful work. The heat is supplied by the combustion of solid, liquid, or gaseous fuel, and we have to consider, first, the most convenient ways of converting the heat into work for different purposes and, second, ways of converting as much as possible of the heat supplied into work. No engine can use all the heat supplied, as some of it must be discharged in the exhaust, and there are other ways in which heat is wasted, as will be seen later.

Engines may be classified broadly as : (1) Steam engines, including steam turbines, in which the heat generated by the combustion of the fuel is used to generate steam at pressures higher than atmospheric, for which boilers are needed. Some of the heat produced is wasted in the flue gases of the boiler, the remainder reaching the engine in the steam produced ; (2) internal combustion engines, in which, as the name implies, combustion of the fuel takes place in the cylinder of the engine. So far the difficulties involved in the use of solid fuel in an engine of this type have not been overcome, although experiments are in progress, and all commercial engines use either liquid or gaseous fuels, such as heavy oil, paraffin, petrol, coal gas, producer gas, or blast-furnace gas.

Heat, its Nature and Effects.—In order to understand how best to utilise heat, we must know something about its nature and effects and how to measure quantities of heat.

Heat was formerly thought to be some kind of subtle fluid, which flowed from hot bodies into colder ones ; but this theory is now no longer accepted, because it was found that heat could be developed to an unlimited extent from cold bodies merely by rubbing them together.

A piece of cold iron can be made red hot by hammering

it. A carpenter's saw, an engineer's chisel, or turning tool, soon get hot when work is done on cutting the material, although they are all quite cold to begin with.

Sir Humphry Davy melted two blocks of ice by rubbing them one upon another, from which he concluded that 'the immediate cause of the phenomenon of heat is *motion*'; and this is now the generally accepted view of the nature of heat.

Still we know that things may be hot without being visibly in motion; hence, if heat is motion, the motion must exist in parts of the body too minute to be seen.

All bodies are assumed to be composed of minute particles called molecules, held together by mutual attraction or cohesion, and these molecules are in a state of continual agitation or vibration. The hotter the body the more vigorous the vibrations of its constituent particles. In solid bodies the vibrations are limited in extent. If this limit is exceeded, owing to addition of heat, cohesion is sufficiently overcome to enable the particles to move about freely and without restriction, and the solid has now become a liquid. On still continuing the heat, further separation of the molecules takes place, cohesion is completely overcome, and they fly off in all directions. The liquid has now become a vapour.

The vapour is formed at a temperature which depends upon the pressure, and the application of further heat while the vapour is in contact with the liquid merely causes further evaporation of the liquid without increasing the temperature. In this case the vapour is known as *saturated* vapour or, if the liquid is water, *saturated steam*. On the other hand, in this case if an attempt is made to cool the vapour some of it will condense without any reduction of temperature. (If saturated steam contains no particles of water in suspension, it is known as *dry saturated steam*. If it consists of a mixture of steam and water particles it is known as *wet steam*.) If, however, the saturated vapour is taken away from contact with the liquid it can be heated to a higher temperature and will not condense until it is

cooled to the temperature of the liquid from which it was generated. It is now *superheated* and has the properties of a gas.

The difference between a vapour such as saturated steam and a so-called 'permanent' gas is that a permanent gas will not condense unless it is cooled to a very low temperature.

The pressure exerted by a gas or vapour on the interior surface of the vessel in which it is confined is due to the collision of the molecules with the sides of the vessel. The greater the intensity of the heat the more violent the impact, and therefore the greater the pressure exerted. This is the condition of things in the interior of a steam boiler.

If a part of the enclosing vessel were movable, it would evidently be pushed backward and outward. This is what happens to the piston of an engine.

From what has been just stated, we see that heat is a form of energy, and that heat and mechanical work are mutually convertible the one into the other. We shall presently show that an exact and invariable relation exists between heat and work.

Temperature.—The *temperature* of a body indicates how hot or how cold the body is, or the *intensity* of the heat of the body.

The *temperature* of a body should be distinguished from the *quantity* of heat in the body. For example, if a cup of water be dipped out of a pailful of water, the *temperature* of the water is the same throughout, but the *quantity* of heat varies as the weight of water in each vessel.

Most British engineers measure temperature in Fahrenheit degrees. On a thermometer graduated in Fahrenheit degrees freezing-point is marked 32° and boiling-point at atmospheric pressure is marked 212°, so that the interval between freezing-point and boiling-point is 180°. Continental engineers and an increasing number of British engineers use the Centigrade scale, on which freezing-point is 0° and boiling-point is 100°.

Useful rules for converting from one scale of temperatures to the other are as follows :

To convert from Fahrenheit to Centigrade, add 40, multiply by $\frac{5}{9}$, subtract 40.

To convert from Centigrade to Fahrenheit, add 40, multiply by $\frac{9}{5}$, subtract 40.

Since heat consists of vibration of the molecules composing a body we may assume that the degree of vibration becomes less as the body loses heat, so that eventually when all the heat has been extracted all motion of the molecules ceases. The loss of heat is shown by a reduction of temperature, and the temperature at which the body has lost the whole of its heat is known as the Absolute Zero of Temperature. This temperature is 273° below the zero of the Centigrade thermometer or 460° below the zero of the Fahrenheit thermometer. The zero of the thermometer is merely a convenient point to measure from and does not indicate the absence of heat in a body.

Absolute temperatures are measured from absolute zero, and the quantity of heat in a body is proportional to its absolute temperature. Hence, to express degrees C. in degrees of absolute temperature, add 273. To express degrees F. in absolute temperature, add 460. Although the amount of heat in a body at 40° C. is not twice the amount of heat in the body at 20° C., the amount of heat it contains at 600° absolute (327° C.) is twice the amount of heat it contains at 300° absolute (27° C.).

Unit of Heat and Unit of Work.—Before quantities of heat can be measured, we must have a unit of heat, just as we require a unit of length, namely, the inch or foot, in order to measure distance ; or the pound or ton, in order to measure weight.

The unit of heat, or thermal unit, is the amount of heat required to raise unit weight of water 1° in temperature. When the unit weight is 1 lb. and the temperature is in Fahrenheit degrees the unit of heat is termed the British Thermal Unit (B.Th.U.). If unit weight is 1 lb. and the temperature is in Centigrade degrees, the unit of heat is

termed the Centigrade Heat Unit (C.H.U.). Where 1 gram is used as the unit weight and the temperature is measured in Centigrade degrees, the unit of heat is termed the Calorie. The relation between these units is 1 C.H.U. $=1 \cdot 8$ B.Th.U. $=$ 454 calories.

Since the amount of heat required to raise 1 lb. of water 1° varies slightly with the temperature, the B.Th.U. is now defined as $\frac{1}{180}$ of the heat required to raise 1 lb. of water from freezing-point to boiling-point at atmospheric pressure (14·7 lb. per sq. in.).

But the all-important point with the engineer is the conversion of heat into *work*. We will therefore now consider what is understood by work, how it is measured, and what the relation is which exists between the two.

By the term *work* in mechanics is understood 'the overcoming of a resistance through a space,' and the amount of work done is measured by the resistance overcome, multiplied by the distance through which it is overcome, the resistance being measured in pounds and the distance in feet.

Thus, if a body weighing 7 lb. be lifted through a height of 3 ft., then the resistance, namely, 7 lb., multiplied by the distance through which it is overcome, namely, 3 ft., is equal to $7 \times 3 = 21$ ft.-lb. of work. Hence, work is measured neither by the pound nor by the foot, but by the product of the two. Thus the *unit of work* is the work done in raising 1 lb. through a vertical height of 1 ft., and is called the *foot-pound.*

Or, since action and reaction are equal and opposite, we may consider the *force* which overcomes the resistance. The work done by a force is measured by the intensity of the force, multiplied by the distance through which it acts, measured in the direction of the force. Thus, as before, in the above example, a force of 7 lb. overcame the resistance due to the weight, and acted through a space of 3 ft., doing thereby $7 \times 3 = 21$ ft.-lb. of work.

Since the unit of work is a product of two numbers, it may be represented by an area, and this is important, for

we intend by-and-by to estimate the work done by an engine from the area of an indicator figure. Thus, if $\frac{1}{8}$ in. be taken to represent pounds on one line, and $\frac{1}{4}$ in. to represent feet on a line at right angles to it, then the unit of work is given by the small cross-lined rectangle, and the 21 ft.-lb. in the above example by the whole rectangle (Fig. 1).

Again, suppose the weight lifted in the previous case to be a vessel containing 7 lb. of small shot, and that the shot should escape by a hole in the vessel at a uniform rate, all the while it is being lifted, until, when a height of 3 ft. is reached, there are no shot left. This result also may be

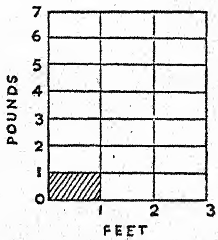

Fig. 1.

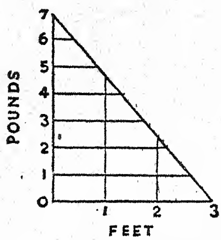

Fig. 2.

well shown by a diagram (Fig. 2), where the weight, varying from 7 lb. to nothing, is given by a diagonal falling from 7 to the zero line of weight. The weight of the vessel is neglected.

The total work done is again given by the area of the figure, and is evidently equal to the distance 3 multiplied by the *mean* weight$=3\times\frac{7+0}{2}=3\times3\frac{1}{2}=10\frac{1}{2}$, or one-half that in the previous case.

It should be noticed that the unit of work has no reference to the *time* taken, for the same amount of work is done in lifting the weight, whether it be done in one second or one hour.

The *power* of an agent is measured by the *rate* at which

it can do work, and depends upon the amount of work done in the unit of *time*.

The unit of power adopted by engineers is the *horsepower.*

A horsepower represents the performance of 33,000 ft.-lb. of work per minute. The addition of the words *per minute* should be particularly noticed.

$$\frac{\text{Work done}}{\text{Time in minutes}}=\text{units of work done per minute};$$

$$\text{and } \frac{\text{Work done}}{33{,}000\times\text{time in minutes}}=\text{horsepower exerted.}$$

Energy is defined as 'capability of doing work.' The steam in a boiler possesses energy even when the stop valve is shut and no work is being done. Some of this energy is converted into work when the steam is allowed to drive the piston of an engine against a resistance. The heat energy in a locomotive boiler is capable of being transformed into the energy of motion of the moving train.

If the brakes are applied to the moving train, then the energy of motion of the train is retransformed into heat, sparks fly from the wheels and rails, and the train is brought to a standstill.

It is a fundamental principle in nature that, just as matter can neither be created nor destroyed, though it may be made to assume different forms, visible or invisible, so energy, whether heat energy or any other, cannot be destroyed. It may take a variety of different forms, but the sum total of the energy remains the same. This principle is called the principle of the *conservation of energy.*

Hence the heat which is carried to the engine in the steam is either transformed into useful work, or it passes away to waste in various ways, and the sum of the heat usefully employed plus the heat which is wasted always equals exactly the heat which was applied.

Mechanical Equivalent of Heat.—Since heat can be transformed into work, and work can be transformed into heat, it is necessary to know how many foot-pounds of

work correspond to one thermal unit. It has been established by experiment that 1 B.Th.U. corresponds to 778 ft.-lb. of work.

$\therefore$ 1 B.Th.U. = 778 ft.-lb. ; 1 C.H.U. = 1400 ft.-lb.

Example 1.—(*a*) In testing an engine, 30 H.P. is absorbed by means of a brake on the flywheel. How much heat is generated per minute ?

Since 30 H.P. = 30 × 33,000 ft.-lb. per min. and 1 B.Th.U. = 778 ft.-lb.

$$\text{Heat generated per minute} = \frac{30 \times 33{,}000}{778} = 1273 \text{ B.Th.U.}$$

$$= \frac{1273}{1{\cdot}8} = 707 \text{ C.H.U.}$$

(*b*) If 40 lb. of cooling water per minute is supplied to the rim, what will be the rise of temperature of the cooling water ?

A rise of temperature of 1° F. will absorb 40 B.Th.U. per minute.

$\therefore$ Rise of temperature necessary to absorb 1273 B.Th.U. per minute $= \frac{1273}{40} = 31{\cdot}8°$ F. (17·7° C.).

Example 2.—A steam engine uses 15 lb. of steam per hour for each horsepower generated. (This is generally spoken of as 15 lb. of steam per horsepower hour.) If the heat required to generate each pound of steam supplied is 610 C.H.U., what fraction of the heat supplied is converted to work ? (This fraction is usually called the *thermal efficiency* of the engine.)

Work done per hour for each horsepower = 33,000 ft.-lb. per minute for 60 minutes = 33,000 × 60 ft.-lb.

$\therefore$ Heat converted to work for each horsepower $= \frac{33{,}000 \times 60}{1400} =$ 1413 C.H.U. per hour.

Heat supplied for each horsepower = 610 × 15 = 9150 C.H.U. per hour.

$\therefore$ Fraction of heat supplied which is converted to work $= \frac{1413}{9150} = 0{\cdot}155$.

The greater part of the remaining heat goes out with the exhaust steam, some of it being lost by radiation, etc.

Specific Heat.—The number of thermal units required to raise 1 lb. of a substance through 1° of temperature is known as the *specific heat* of the substance. Thus if we say that the specific heat of some metal is 0·12 we mean that it requires 0·12 C.H.U. to raise 1 lb. of it by 1° C. or

0·12 B.Th.U. to raise 1 lb. of it by 1° F. The specific heat of water will obviously be unity.

The specific heat of bodies varies very considerably, as will be seen from the following table :

I. Table of Specific Heats

Cast iron	=0·12
Steel	=0·12
Copper	=0·09
Brass	=0·09
Lead	=0·03
Mercury	=0·033
Zinc	=0·09
Petroleum	=0·51

Water has the highest specific heat of any substance (except hydrogen), and the metals have the lowest. In other words, it takes more heat to raise the temperature of a given weight of water 1° than to raise the same weight of any other substance 1°. The specific heat of water is 1. The specific heat of steel by the table is 0·12, or about $\frac{1}{8}$, that is to say, the quantity of heat which would raise 1 lb. of steel through 1° F. would only raise the temperature of 1 lb. of water through about $\frac{1}{8}$° F.

The heat gained by a substance due to a given rise of temperature is given by

$$\text{Heat gained} = \text{weight} \times \text{specific heat} \times \text{rise of temperature.}$$

Thus the heat required to raise the temperature of 25 lb. of brass from 30° C. to 98° C. would be $25 \times 0{\cdot}09 \times (98-30)$ $=153$ C.H.U.

CHAPTER II

TRANSFER OF HEAT

When bodies of unequal temperature are placed near each other, the hot body tends to part with its heat to the colder body until the temperature in each is equal ; and when there is no tendency to a transfer of heat between them they are said to be of equal temperature.

The transfer of heat from one to the other may take place in any of the following ways : namely, by radiation, conduction, or convection.

The heat from the surface of the burning coal in a furnace is transferred to the crown and sides of the furnace by radiation ; it passes through the furnace plates by conduction, and the water is heated by convection.

Radiation.—Radiant heat passes from one body to another practically instantaneously without any intervening substance being required. Substances which are transparent to light usually allow radiant heat to pass freely through them, even when they are very poor conductors. This is one reason why radiant heat is so important in boilers, since it will pass through a thin film of gas clinging to the plates or tubes which offers considerable resistance to the passage of heat by conduction.

Conduction.—The process by which heat passes from hotter to colder parts of the same body, or from a hot body to a colder body in contact with it, is called *conduction.* A bar of iron having one end placed in the fire soon becomes hot at the other extremity, the heat being conducted from particle to particle throughout its entire length.

A piece of burning wood can be held with the hand close to the burning part. Evidently, therefore, some bodies conduct heat much more readily than others.

If a piece of clean paper be pasted on the bottom of a

copper kettle containing water, and the kettle be placed on a bright fire or over a strong gas flame, the water will soon be warmed, but the paper will not be charred in the least; the reason of this being that the heat is so rapidly conducted by the copper to the water. Bodies which conduct heat readily are called good conductors; those which conduct heat slowly are called bad conductors.

'Non-conducting' materials do not really exist, but the term is usually applied to those materials which have very low conductivity. Such materials are used to cover the external surfaces of boilers, steam pipes, and steam-engine cylinders to reduce loss of heat by radiation. Air is an extremely poor conductor of heat, so that 'lagging' materials which contain a large number of fine air cells (to prevent circulation of the air) usually give the best results.

Liquids and most gases are very poor conductors of heat, and it is a difficult and very slow process to heat them by conduction, but if they are allowed to circulate they may be readily heated by convection.

The following table shows the effect of lagging a steam pipe with different thicknesses of magnesia lagging:

TABLE II

Temporary excess of pipe above room (degrees F.)	Loss in B.Th.U. per sq. ft. per hour			
	Bare pipe	Magnesia lagging		
		1 in.	1½ in.	2 in.
350	1200	178	137	112
400	1550	208	156	128
450	1950	234	180	148
500	2400	260	200	165
700	4800	378	287	238

Consider a pipe 6 in. diameter and 40 ft. long containing steam at 500° F. The total surface of the pipe is 63 sq. ft. and with an uncovered pipe the loss per hour would be

$2400 \times 63 = 150,000$ B.Th.U. This is equivalent to an extra 15 lb. of coal per hour burned in the boiler. In a year of 2000 working hours the extra coal burned would be 13·4 tons, costing £20 at 30*s*. per ton. Lagging the pipe with $1\frac{1}{2}$ in. of magnesia would reduce the annual loss to less than £2, the cost of lagging being saved in less than 12 months.

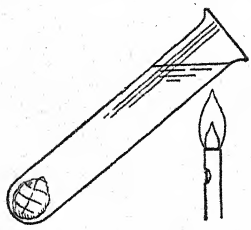

FIG. 3.

Convection.—C o n v e c t e d or carried heat is that which is transmitted from one place to another by currents. The following experiment will clearly show that water is a bad conductor, and the necessity therefore of heating it by some other method than conduction. Take a test-tube nearly full of cold water, and hold the tube with the upper surface of the water against a flame, as shown in Fig. 3. The water will soon boil at its upper surface, while the temperature of the water in the bottom of the tube is not appreciably changed ; for, if a piece of ice be placed in the bottom of the tube, it will remain unmelted. If, however, the heat be applied at the bottom of the vessel, Fig. 4, the heated lower layers, becoming less dense, rise towards the surface, while the cold upper and denser layers fall and thus circulating currents are set up which can be very plainly seen by dropping a little bran into the water, and which soon result in the water being heated throughout.

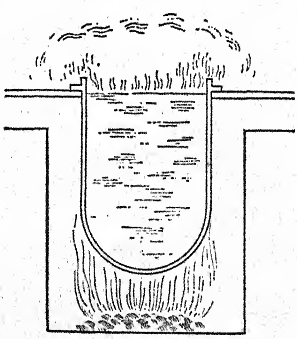

FIG. 4.

In the arrangement shown in Fig. 4 the circulation is likely to be more or less confused, interference of currents takes place, and if the heat is intense there is violent

agitation at the surface, causing the water to 'boil over.'

If, however, an inner vessel having openings at the bottom and top be placed so as to leave an annular water-space as shown in Fig. 5, then the upward and downward currents will be separated, there will be free circulation without interference of currents, and the boiling will take place much more smoothly and efficiently.

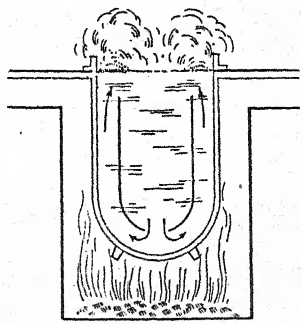

FIG. 5.*

Similarly, if a U-tube is taken and heat applied to one leg only, circulation is immediately set up, the heated and less dense water in the hot leg rising while the colder and denser liquid in the other leg flows downwards, following up the moving water and completing the cycle.

As soon, however, as the water is heated throughout to boiling-point, steam-bubbles are rapidly formed, and the circulation now continues much more vigorously. This is due to the effect of the velocity with which the steam-bubbles tend to rise in the denser surrounding water, and to the resistance offered by the water to the upward motion of the steam-bubbles. The sum of these resistances in the column of water is the measure of the force setting up an upward flow of the water. The rate of circulation produced in this way may be very great.

Figs. 6 and 7 are modifications of the U-tube, and they are devices for increasing heating surface, while retaining the advantages of free circulation.

A free circulation is important in steam boilers, (i) to maintain the boiler at a uniform temperature, so as to prevent unequal expansion in the various parts of the boiler, especially in boilers having thick plates; and

* This and the four following figures are kindly lent by Messrs Babcock and Wilcox.

(ii) to facilitate the escape of steam from the heating surface as soon as it is formed, so as to prevent overheating

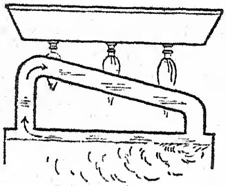

FIG. 6.

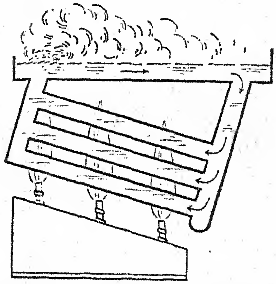

FIG. 7.

of the plate, which would quickly occur unless the plate is maintained in constant and immediate contact with water.

CHAPTER III

EFFECTS OF HEATING AND COOLING

All bodies expand by the action of heat. Numerous examples of the application of this law of expansion of metals will occur to students of engineering. Thus the bars of boiler furnaces are left free at the ends to enable them to expand. Boiler plates are riveted with red-hot rivets which cool and contract and draw the plates together at the joint with great force. In laying railways, a small space is left between successive lengths of rail ; and the bolt-holes by which they are secured to the fish-plates are elongated. Tires of wheels are fitted on when red hot, and as they cool they contract and grip the wheel with great firmness. Cranks are ' shrunk on ' crankshafts in a similar way. The walls of buildings which bulge out in the centre have been drawn back into position by passing iron bars through the walls from side to side of the building. They are screwed at the end with nuts and have large plate washers. The bars are heated inside the building, and the nuts are tightened up. On cooling, the bars contract and draw the bulged walls together. Steam pipes which are rigidly secured between two supports should be fitted with an expansion joint or connection (see Fig. 8).

Engine cylinders, which are heated to the temperature of the steam, instead of being rigidly bolted down on a horizontal bed-plate, are frequently secured by the front face, the rest of the cylinder overhanging the bed of the engine. A small space is allowed between the crank bosses and main bearings of engines having cast-iron bed-plates, to allow of expansion of the crankshaft in case of hot bearings, etc. If glass is heated or cooled suddenly, it is very liable to crack, because glass conducts heat slowly, and the two sides of the glass are unequally heated, and therefore

unequally expanded : hence the fracture. The same thing is liable to occur in steam cylinders, which should always be carefully warmed by opening the stop valve a little while the steam is being generated, and blowing gently through with steam for some time before starting the

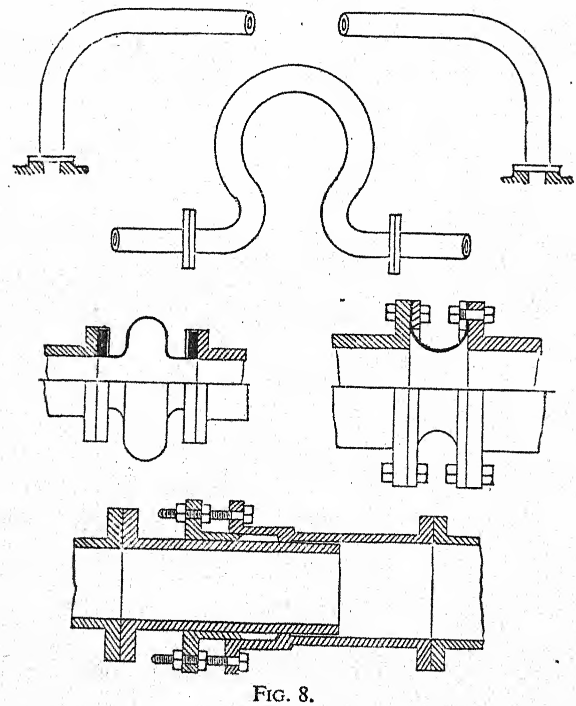

FIG. 8.

engines, and thus bringing the cylinders and jackets gradually up to the working temperature.

Steam boilers of the 'tank' type also require great care for similar reasons. They should not be hurriedly heated or cooled, and all sudden changes of temperature should be avoided ; otherwise, unequal expansion and contraction will take place, resulting in leakages.

Less harm is done to a boiler of this type by steaming steadily for a length of time than by repeatedly getting up steam and drawing the fires, which bring about repeated expansions and contractions of the boiler.

The force exerted by heat in expanding a bar of metal is the same as would be required to stretch it to the same extent by mechanical means.

		Per degrees F.
Coefficient of expansion for iron and steel		$=0{\cdot}00000683$
,, ,, ,,	copper	$=0{\cdot}00000956$

That is to say, if a cold wrought-iron pipe at 38° F. be gradually warmed up by steam, and eventually filled with steam at 100 lb. boiler pressure, or 338° F., this pipe will have expanded in length by an amount $=(338-38)\times 0{\cdot}00000683=0{\cdot}002=\frac{1}{500}$ of its original length, or it has expanded about 1 in. per 42 ft. of length.

In the case of steel, a stress of 30,000 lb. per sq. in. of section will stretch or compress a bar or pipe by $\frac{1}{1000}$ of its length. A pipe 6 in. diameter and $\frac{1}{8}$ in. thick will have a sectional area of 2·35 sq. in., so that a force of 70,500 lb. (31·5 tons) would change its length when cold by the above amount. A rise of temperature of 146° F. would cause the same increase in length, and the force the pipe would exert on rigid supports which prevented expansion would be 31·5 tons in this case. The necessity for allowing free expansion will be obvious.

Fig. 8 gives examples of various methods of providing for expansion of pipes due to heat, so as to prevent the undue and even dangerous straining which occurs when pipes are fixed between rigid supports, with no provision for expansion and contraction.

Absolute Pressure.—A pressure gauge attached to a boiler, or a receiver of compressed air, never shows the true pressure of the steam or gas, even when it has no errors. All it shows is the *difference* between the internal and external pressures. As it is this difference which is tending to burst the vessel the indication of the gauge is useful in this respect, but when we are dealing with the

expansion of steam or gases we must use the true pressure, usually known as the *absolute* pressure.

As the external pressure is usually that due to the atmosphere, this pressure must be added to the gauge pressure in order to obtain the absolute pressure. Since very few gauges registering fairly high pressures are accurate to within less than $\frac{1}{2}$ lb. per sq. in., it is usually sufficiently accurate to add 15 lb. per sq. in. to the gauge reading, thus

Absolute pressure

$$= \text{Gauge pressure} + 15.$$

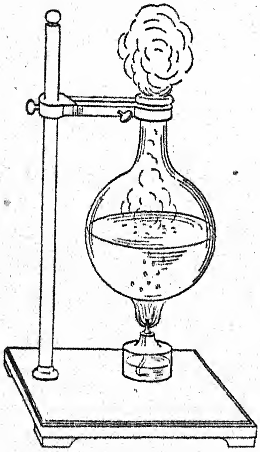

FIG. 9.

Application of Heat to Water.—Water is a compound substance, consisting of hydrogen and oxygen chemically combined in the proportion of two volumes of hydrogen to one volume of oxygen, written in chemical symbols H_2O.

When water is subjected to the action of heat it is converted into *steam*, which is water in the state of a vapour.

Though a change thus takes place in the physical condition of the substance, the chemical composition of the steam is in no way different from that of the water from which it is generated.

Boiling.—If heat be applied to the bottom of a vessel, as in Fig. 9, the air contained in the water will first appear as little bubbles which rise to the surface. Then the water immediately in contact with the source of heat will be converted into steam. The steam will form as bubbles on the bottom, and these will rise through the liquid ; but at the commencement of the operation they will at once be condensed by the cold upper layers of water. The condensation of the bubbles of steam is the cause of the 'singing' of the water before boiling. Finally, the water

becomes heated throughout until it reaches a temperature of 212° F. under *the pressure of the atmosphere*, when the bubbles rise to the surface and boiling begins.

It should be particularly noted that the temperature at which boiling takes place depends upon the pressure on the liquid, and that for every different pressure there is a fixed temperature at which boiling takes place, so that water has an indefinite number of boiling points.

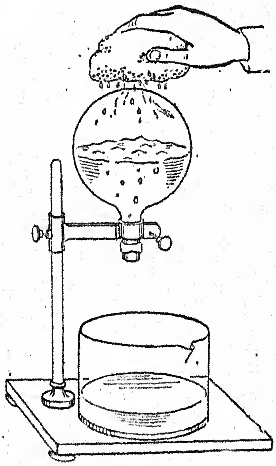

FIG. 10.

An experiment illustrating boiling at a low temperature will be understood by reference to Fig. 10. Water is boiled in a glass flask as in Fig. 9. When the water has been boiling a little time, and all the air is expelled, the heat is removed, and the flask is closed by a cork, turned upside down, and placed on the stand as shown. Meantime the water has, of course, ceased to boil. If now cold water be poured gently on the flask, the steam which occupies the space above the water will be condensed, *the pressure on the water will therefore be reduced*, and the water will again boil vigorously, although the temperature of the water has by this time fallen considerably below 212° F.

Similarly, owing to the reduced pressure of the atmosphere on the tops of high mountains, boiling water is not sufficiently hot to cook food. On the other hand, the temperature of boiling water at the bottom of deep mines is higher than at the surface.

The boiling temperatures for water under varying pressures are given in Table III, p. 37. For instance, under a pressure of 5 lb. per sq. in. abs. the boiling-point is 162·3° F. (72·4° C.). while at 250 lb. per sq. in. abs. it

is 401·2° F. (205·1° C.). This dependence of temperature on pressure is used in process work by selecting a steam pressure which will give the required high or low temperature for the particular process.

The presence of solid bodies such as salt dissolved in the water raises the temperature of the boiling-point. Thus the boiling-point of sea water under atmospheric pressure is 213·2° F.

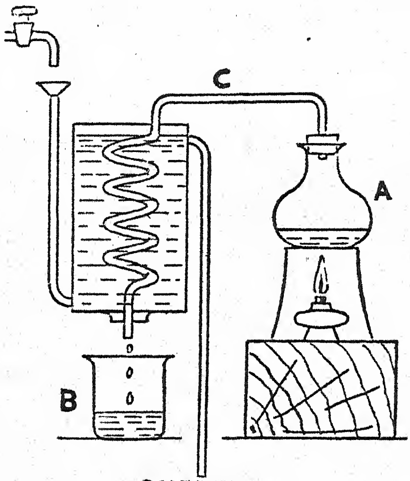

FIG. 11.

Condensation of Steam—Vacuum.—Steam is water vapour, and when the steam is cooled, it again returns to the liquid state.

Thus, let a flask A contain a known weight of water. Fit a cork and glass tube to it as shown, and connect with a spiral tube surrounded by flowing cold water ; let the lower end of the tube pass into a vessel B. Boil the water in A. It will pass off as steam by the tube C to the spiral ; and if the spiral be surrounded by a stream of cold water, the steam will be condensed to water, which will drop from the end of the tube.

At the end of the operation the loss of weight by A is equal to the gain by B. This illustrates the process of distillation, and by this method pure water may be obtained from water containing impurities.

Advantage was taken by the early engineers of the property possessed by steam of being easily condensed. They valued steam not so much for its own sake, but because by condensation they were able to call to their aid the pressure of the atmosphere in the performance of work.

A *vacuum* is literally an *empty space*—that is, a space absolutely free from air or vapour of any kind capable of exerting pressure.

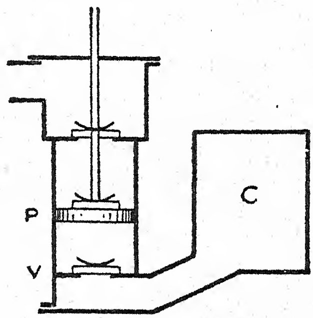

FIG. 12.

Vapour arises from water at *all* temperatures, and exerts an appreciable pressure. And the lowness to which the pressure can be reduced in condensers depends on the temperature of the condensed steam, and this temperature in practice can seldom economically be reduced below about 102° F., at which temperature the vapour of water exerts a pressure of 1 lb. per sq. in.

Nearly all steam contains air which was dissolved in the feed water entering the boiler, and as this air cannot be condensed it must be removed from the condenser by some form of *air-pump*, an old type of which is shown in Figs. 12 and 97.

When the plunger or pump bucket P is lifted, the valve V will lift by virtue of the difference in pressure on the two sides of the valve. Assuming that we could obtain a perfect vacuum in the pump chamber, yet the pressure per square inch in the condenser C can never fall below that necessary to lift the valve V. In newer types of air-pump this necessity does not exist (see Chapter X).

Experiment.—Take a thin tin cylinder closed at both

ends, having a tap, t, at one end. Pour a little water into the cylinder by the tap. The vessel now contains air and water. Boil the water till the steam escapes from t and has driven most of the air out. Now the vessel contains steam and very little air. Close the tap and pour cold water on the vessel. The steam is immediately condensed to water; and since water occupies only about $\frac{1}{1650}$ of the space of the steam at atmospheric pressure, a partially empty space has been formed inside the vessel, and the external pressure of the atmosphere will collapse or crush the vessel.

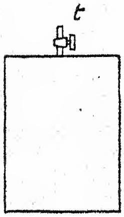

FIG. 13.

If the cylinder had been made strong enough to resist the excess of external pressure over internal pressure, and a tube had been led from the cylinder into water some depth below it, then the water would be forced up the tube into the cylinder by the pressure of the atmosphere, till the pressure on the inside of the cylinder is the same as the atmospheric pressure outside. Here, then, useful work would be done in lifting water from a low level to a higher level, and this was the principle of the early pumping engines as made by Savery. Again, if the top of the cylinder had been movable, then it would act like a piston, and be forced towards the bottom. This was the principle of Newcomen's engine, which was called the 'atmospheric' engine, because the work was really done by the atmosphere on the piston after a vacuum had been formed in the cylinder by the condensed steam.

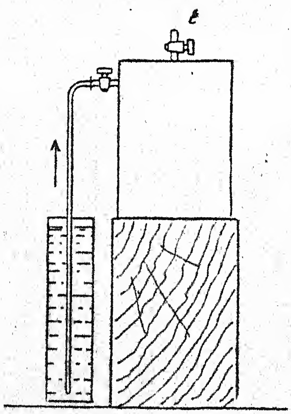

FIG. 14.

Savery's engine, having drawn its water from a low level

into a chamber, as previously explained, delivered it to a still higher level above the chamber by introducing steam to the same chamber, and forcing the water up the delivery pipe to a higher level by the pressure of the steam upon the surface of the water.

This principle has been again revived in the Pulsometer and similar pumps.

The Pulsometer is illustrated in Fig. 15, and consists of a single casting called the body, composed of two chambers A, A joined side by side with tapering necks, the two passages terminating in a common steam chamber, wherein the ball valve I is fitted so as to oscillate between the seats at the opening to each chamber.

Between the chambers A, A and the suction pipe C are the suction valves E, E as shown. A discharge chamber common to both working chambers, and leading to the discharge pipe G, is also provided, and this contains delivery valves F, F. The air-chamber B communicates with the suction.

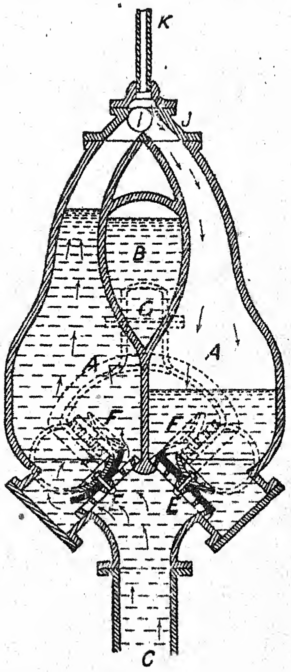

FIG. 15.

The pump being first filled with water, steam is admitted by the steam pipe K, passes down that side of the steam neck which is left open to it by the position of the steam ball, and presses upon the surface of the water in the chamber, depressing it without agitation of the water, and therefore without much condensation of steam, and forcing the water up the delivery pipe. The moment that the level of the water is as low as the horizontal orifice which leads to the discharge pipe, the steam flows through with a certain amount of violence, causing agitation and instan-

taneous condensation of the steam in the chamber, a vacuum is formed, and the steam ball falls over, closing the mouth of the chamber. This prevents further admission of steam, and allows the vacuum to be completed ; meantime water rises through the suction valve, and rapidly fills the empty chamber.

The same operations are repeated in the other chamber, and proceed alternately in the two chambers, one delivering while the other is being filled.

Newcomen's 'Atmospheric' Engine.—This engine was devised by Newcomen, a blacksmith of Dartmouth, in 1705, and though of the crudest design and construction, this type of engine served a useful purpose for many years as a pumping engine for mines, until it was displaced by the greatly improved engine introduced by James Watt.

The Newcomen engine is illustrated in Fig. 16. Steam was admitted in the first place from the boiler B to the cylinder D. The steam below the piston was only at about atmospheric pressure, and the piston, being in equilibrium (having equal pressure above and below it), was raised from the bottom to the top of the cylinder by the greater weight of the pump-rod H suspended from the opposite end of the main beam G, and acting as a counterpoise.

When the piston reached the top of the cylinder the steam was shut off, and a jet of cold water was sprayed into the cylinder, condensing the steam, and thereby forming a partial vacuum under the piston. The atmospheric pressure then forced the piston downwards, and through the medium of the beam the pump-rod was raised. On the next steam admission the water in the cylinder was expelled, and the operations above described repeated.

It was while experimenting with a model of this engine that James Watt, in 1763, discovered how large a waste of steam was going on in the cylinder, owing to condensation of the steam through contact with the cold wet walls of the cylinder.

It was this waste by condensation that led Watt to the invention of the *separate condenser*, the steam being passed from the cylinder into a second chamber called the con-

denser, where it might be condensed by contact with cold water without the need of cooling the cylinder itself (see Fig. 103).

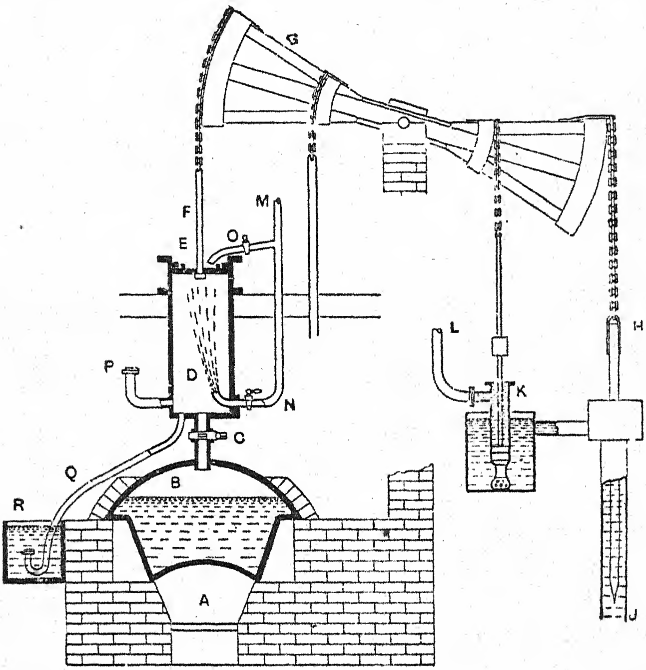

FIG. 16.

A = the boiler furnace
B = the steam boiler
C = the steam valve
D = the engine cylinder
E = the piston
F = the piston rod
G = the main beam
H = the heavy pump rod
J = the mine pump
K = pump for condensing water
L = pipe leading to condensing water-tank
M = condensing water-pipe
N = injection cock to cylinder
O = water-tap to top piston
P = relief or shifting valve
Q = eduction pipe with non-return valve at end
R = feed-water tank

This invention by Watt of the separate condenser, though apparently so simple, was the secret of the great success which followed the steam engine from this time forward.

CHAPTER IV.

ACTION OF HEAT IN THE FORMATION OF STEAM

The action of heat in the formation of steam from water may be illustrated by the following diagrams.

(1) Let the cylinder (stage 1, Fig. 17) contain 1 lb. of water at 32° F., and let the pressure of the atmosphere be

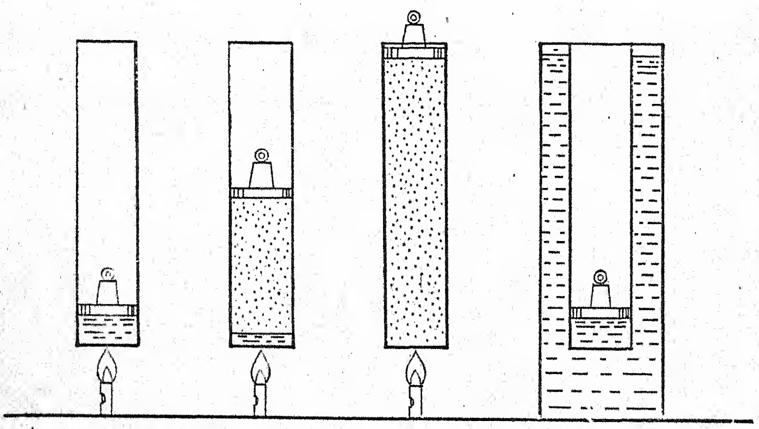

FIG. 17.

represented by a weighted piston. Then, if heat be applied to the water, the temperature will rise higher and higher, though the piston will remain stationary, except for the small expansion of the water, until the temperature of the water reaches 212°.

(2) On continuing the heat the water shows no further increase of temperature by the thermometer, but steam begins to form and the piston commences to ascend in the cylinder (stage 2), rising higher and higher as more and

more steam is formed, until the whole of the water is converted into steam. In stage 1 the steam did not begin to form until the temperature reached 212°. Evidently, therefore, this is the lowest temperature at which steam can exist under atmospheric pressure.

Again, in stage 3, as soon as the last drop of water disappears, we have 1 lb. of steam occupying the least possible volume at the given pressure ; the steam in this condition is termed dry *saturated* steam.

(3) If the heat is continued the steam will become *superheated*—that is, its temperature will rise above that of saturated steam, and the piston will continue to rise.

(4) If the steam be surrounded by a vessel containing an indefinite supply of cold water (stage 4), then the heat will be extracted from the steam by the surrounding water, and the steam will be condensed to water, the same in every particular as to weight and properties as the water with which we started ; and if the temperature of the water is now the same as its temperature before starting, then the whole heat taken away when the steam is condensed is equal to the whole heat added during the operation. The series of changes have, therefore, been brought about by the addition or subtraction of heat only.

We have so far been content with a general statement of the action of heat in the formation of steam ; we will now consider what *quantities* of heat are required to perform the several stages of the operation.

Work done by Steam during Formation.—Referring to Fig. 18, let 1 lb. of water at 32° F. be contained at the bottom of a cylinder 1 sq. ft., or 144 sq. in., in sectional area. Then, first to find the height of the water in the cylinder ; since the area of the vessel is 1 sq. ft., and the weight of 1 cu. ft. of water is 62·5 lb.,

62·5 lb. of water will stand 1 ft. high,

1 lb. " " $\frac{1}{62 \cdot 5}$ ft.

$=0 \cdot 016$ ft.

Let the pressure of the atmosphere be represented by a piston resting on the surface of the water loaded with a weight of 14·7 lb. per sq. in.

The area of the piston being 1 sq. ft., the total weight on the piston is therefore $14 \cdot 7 \times 144 = 2116 \cdot 8$ lb.

(1) On applying heat to the water, it will at first gradually rise in temperature from 32° to 212° before evaporation commences, as explained on p. 26.

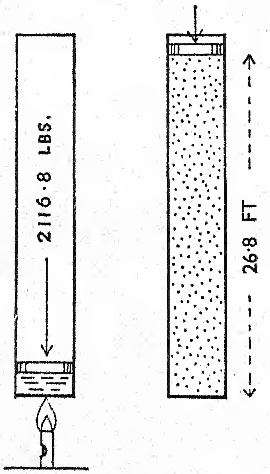

FIG. 18.

Then, $212 - 32 = 180$ = the number of heat units required to raise water from 32° to boiling temperature at atmospheric pressure, and this represents the heat units expended in stage 1, Fig. 18.

(2) Steam now begins to form and the piston to rise ; and, on continuing the heat, the water is eventually all converted into steam at 212°, and the piston continues to rise till the steam occupies a volume, under the pressure of the atmosphere of 26·8 cu. ft. (see Tables, p. 37).

The heat expended in evaporating the 1 lb. of water at 212° into 1 lb. of steam at 212° is found to be 967 B.Th.U. Hence the total heat required, first to raise water from 32° to 212°, and then to convert it into steam at the same temperature under atmospheric pressure, $= 180 + 967 = 1147$ B.Th.U. Now, in stage 1, Fig. 17, it is quite evident how the heat has been expended, namely, in raising the temperature of the water ; but in converting the water into steam, though 967 units of heat have been expended, there is no increase in temperature, and it is not at first quite clear what has become of this heat ; hence it was called latent or hidden heat. What has become of this heat, however, will be understood from the following explanation :

It will be noticed that in this operation two things have happened: firstly, the water has all been converted into steam, which occupies a greatly increased volume (1675 times, at atmospheric pressure) as compared with the water from which it was generated; and, secondly, the piston has been raised from the surface of the water in stage 1 to that of the steam in stage 3. The heat, usually called latent heat, has been expended, then, in two ways: firstly in overcoming the internal molecular resistances of the water in changing its condition from water to steam; and, secondly, in overcoming the external resistance of the piston to its increasing volume during formation.

The first of these effects of 'latent' heat is called *internal* work, because the changes have been wrought within the body itself; and the second is called *external* work, because the work has been done on bodies external to itself; and these two kinds of work must be carefully distinguished. The first represents energy contained *in* the steam; the second represents energy which has passed out of it, having been expended in doing work on the piston.

We will now consider what share of the heat has been expended on each operation respectively.

The heat expended in doing the external work of raising the piston under a pressure of 2116·8 lb. through a height of 26·8 ft. $=2116{\cdot}8\times26{\cdot}8=56{,}600$ ft.-lb.; or $56{,}600\div778$ $=72{\cdot}8$ B.Th.U.

Now, the total heat applied to the water, as we have seen, is 1147 units; and we have so far accounted for $180+73$ $=253$ units, leaving a difference of $1147-253=894$ units, and this difference represents the heat absorbed in doing the internal work of converting the water into steam.

The distribution of the heat may be summarised as follows:

	B.Th.U.
(1) In raising temp. of water from 32° to 212° =	180
(2) In overcoming internal resistance =	894
(3) In raising piston against external resistance =	72
Total heat =	1146

Now, the external work done per lb. of steam during its formation may be represented by an area. For the pressure P per square foot multiplied by the area of the piston in square feet gives the load on the piston, and this multiplied by the height l through which the piston moves in feet gives the work done; or

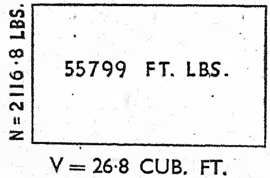

FIG. 19.

$$\text{External work} = P \times a \times l.$$

But $a \times l = v$ = the volume occupied by the 1 lb. of steam; therefore

$$\text{External work} = P \times v.$$

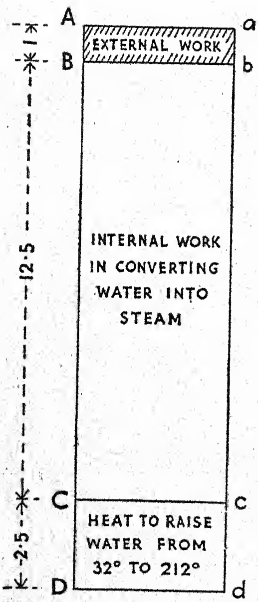

FIG. 20.

If, then, a rectangle be constructed, as in Fig. 19, having one side=P, and an adjacent side=v, to any convenient scale, the area of the rectangle will equal the work done.

Similarly, the proportion which the heat converted into external or useful work bears to the whole heat expended may be shown by the aid of rectangular areas.

From the above summary of results we see that the ratio of the thermal units expended as described is as 180 : 894 : 72; or, dividing each of the numbers by 72, we have 2·5 : 12·4 : 1.

Draw the rectangle ABba (Fig. 20), making AB=pressure and Bb=volume to any scale to represent the external work done by the steam. To the base Bb add the rectangle BCcb=12·4 times the rectangle Ab. This is done by making BC=12·4 times AB. Make also CD=2·5 times

AB and complete the rectangle. Then the total heat required to heat 1 lb. of water from 32° to 212°, and to convert it into steam at the same temperature, is given by the rectangle AD*da*, and the share of this which goes to perform useful work is represented by the remarkably small area given by the rectangle AB*ba*.

Thus, in such an engine as this, taking steam at full pressure throughout the whole stroke, only $\frac{1}{16}$ of the heat is usefully employed, while the remainder escapes into the air or condenser in the exhaust steam, except the small part which is wasted by radiation and conduction.

Hence, for every 16 lb. of coal consumed, the heat from 1 lb. only is converted into work, or $\frac{1}{16} \times 100 = 6{\cdot}25$ per cent. And this is better than would be the case in practice under the same circumstances, because we have neglected the many sources of loss which will be described hereafter.

We may now consider the effect of using steam at a higher pressure than that of the atmosphere. Take, for example, steam at 100 lb. per sq. in. abs.

The external work done by 1 lb. of steam at 100 lb. pressure per square inch absolute, having given that 1 lb. of steam at 100 lb. pressure occupies 4·45 cu. ft., is found as follows:

$$\mathrm{P} = 100 \times 144 = 14{,}400 \text{ lb.}$$
$$\text{and } v = 4{\cdot}45 \text{ cu. ft.}$$

Then total external work of steam during formation $= \mathrm{P} \times v$

$$= 14{,}400 \times 4{\cdot}33 = 64{,}100 \text{ ft.-lb.}$$

Comparing this with the external work done by 1 lb. of steam at atmospheric pressure, we have

			external work in ft. lb.
1 lb. steam	at 100 lb.	pressure	=64,100
1 lb.	„ 14·7	„	=56,600

and these numbers only differ by 13 per cent.

From this we see that, when steam is *not used expansively*—that is, when it is supplied at full pressure throughout the stroke—1 lb. of high-pressure steam is not capable of doing much more useful work than the same *weight* of low-pressure steam.

In comparing the work done by high- and low-pressure steam, it will be noticed we have taken the work done by equal *weights* and no expansion. The same would not be true of equal *volumes*, for evidently if the cylinder were supplied with high-pressure steam, it would do more work on the piston than the same volume of steam at a lower pressure ; but then there would be a proportionally greater weight of steam used, and, therefore, a greater quantity of fuel consumed ; and the object of the engineer is to get the greatest amount of work from the least consumption of fuel. Thus, if a cylinder is filled at each stroke with steam at 100 lb. pressure per square inch throughout, then, assuming there is no back pressure, this steam would do twice as much work as steam at 50 lb. ; but the weight of each cylinder full at 100 lb. pressure is approximately twice that of the cylinder full at 50 lb. Hence, though we have done twice the work, we have used twice the weight of steam, and, therefore, weight for weight, the work done in both cases is equal.

To find the work done per pound of steam during formation, without expansion, at any given absolute pressure per square inch p :—Find by Table III (p. 37) the given pressure, the volume v per pound in cubic feet ; then $p \times 144 \times v =$work done.

Example.—Find the external work done per 1 lb. of steam at 60 lb. pressure absolute ; then by Table III, volume per lb. of steam at 60 lb. pressure$=7{\cdot}16$ cu. ft., and $60 \times 144 \times 7{\cdot}16 = 61{,}900$ ft.-lb. per lb.

To find the weight of steam required per horsepower per hour : Divide work done per horsepower per hour by work done per pound of steam.

The work done per horsepower per hour$=33{,}000 \times 60$ $=1{,}980{,}000$ ft.lb. The work done per pound of steam at 100 lb. pressure absolute without expansion$=64{,}100$ ft.-lb.

Therefore, the number of pounds of steam required per horsepower per hour under the above conditions

$$= \frac{1{,}980{,}000}{64{,}100} = 30{\cdot}9 \text{ lb.}$$

Heat rejected by Steam to Condenser.—When steam is condensed, the heat rejected by it to the condensing water is not always the same, but depends upon the conditions under which it is condensed. If it is condensed under the same constant pressure at which it was formed, the heat given out will be the same as the total heat supplied; in other words, the heat rejected is the same as its total heat of formation; but if it be condensed under any other conditions, the heat rejected by the steam to the condensing water will be different. This statement may be illustrated by taking three cases:

1*st case*.—Referring again to Fig. 17, stage 4, suppose that when the last particle of water is evaporated, we now commence to cool down the cylinder till the steam is condensed, and converted finally to water at 32°, the piston having fallen to its first position. Now, it will be evident that just as the formation of steam took place under the constant pressure of the weighted piston, so condensation has here been carried on under the same constant pressure, and the whole of the process of formation has been exactly reversed.

FIG. 21.

Hence, heat rejected by water in falling from 212° to 32°=180 B.Th.U.; heat rejected by steam= heat absorbed in internal work=894 B.Th.U.; and, lastly, heat expended in raising piston which has been restored to steam by piston compressing it back to original volume as water=72 units; and, therefore,

$$\begin{aligned} \text{Heat rejected} &= 180+894+72 \\ &= 1146 \text{ B.Th.U.} \\ &= \text{total heat supplied.} \end{aligned}$$

2*nd case*.—Suppose, in Fig. 21, that, when the cooling commenced, the piston had been secured so that it could not fall as the volume of the steam decreased. Then evidently the heat rejected would be less than in the previous case by the amount of work done on the steam by the falling piston under atmospheric pressure; or,

$$\text{Heat rejected} = \text{total heat} - \text{external work}$$
$$= 1146 - 72$$
$$= 1074 \text{ B.Th.U.}$$

for this particular case.

This corresponds to the amount of heat rejected when the steam is exhausted to a condenser without back pressure.

3rd case.—Suppose now that the steam is exhausted into a condenser against a back pressure of say one-third of the pressure of the atmosphere. Then the effect is the same as though, when the piston had arrived at the extreme height due to the volume of 1 lb. of steam at 212° under the pressure of the atmosphere, the piston is secured, the weight representing the atmospheric pressure slipped off, and a weight one-third this size placed on the piston (Fig. 22). Then, when the steam has been cooled till it only exerts a pressure of 5 lb. per sq. in., the piston will begin to fall, and, on continuing the cooling operation, the steam is condensed to water, and the water falls to 32°. Here the stages during the formation of steam have been reversed, except that the work done on the steam by the falling piston will be only one-third of that done on the piston by the steam ; hence

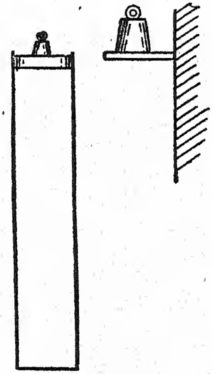

FIG. 22.

$$\begin{aligned} \text{Heat rejected} = \text{heat of water from } 212° \text{ to } 32° &= 180 \\ + \text{internal latent heat} \quad &= 894 \\ + \tfrac{1}{3} \text{ external work} = \tfrac{1}{3} \text{ of } 72 \quad &= \ \ 24 \\ \hline &\ 1098 \end{aligned}$$

We shall now be able more fully to appreciate the meaning of the following definitions :

Sensible heat is the heat added to the water which changes its temperature, and the term is used to denote the heat

required to raise the temperature of 1 lb. of water from 32° to the given temperature. Thus for water at boiling temperature under atmospheric pressure the sensible heat $=212-32=180$ B.Th.U.

If the temperature of the water to begin with is, say, 50° F. instead of 32°, then the number of thermal units required to raise water at 50° to water at $212°=212-50=162$.

The latent heat of steam is defined as the amount of heat required to convert 1 lb. of water at a given temperature into steam at the same temperature.

The *total heat of evaporation* is the sum of the latent and sensible heats and is defined as the quantity of heat required to raise 1 lb. of water from 32° F. to the temperature of evaporation, and to convert it into dry saturated steam at that temperature.

When the product of evaporation is not dry steam, but a mixture of steam and water, as is often the case, then the total heat of 1 lb. of the mixture of steam and water is less than the total heat of the same weight of pure steam. It is the heat required to raise the whole 1 lb. weight of the mixture to the evaporation point plus the latent heat of the proportion of it converted into steam. Thus let s be the sensible heat of the steam and x the dryness fraction of the steam, that is, the proportion of dry steam per pound, then

$$\text{Total heat}=s+x\text{L}$$

where L=the latent heat of the steam at the given pressure as shown in Table III (p. 37).

The *internal latent heat* is that portion of the latent heat which is contained in the 1 lb. of steam after formation; thus—

Internal latent heat=(latent heat)—(heat absorbed in doing external work during formation).

The *internal or intrinsic energy* of the steam includes the internal latent heat and the sensible heat reckoned from 32°; or—

Intrinsic energy=(total heat)—(heat absorbed in doing external work during formation).

From the Table III, it will be evident that the sensible heat increases as the temperature of the steam increases, while the latent heat decreases as the temperature increases.

Example 1.—Find the internal energy of 1 lb. of dry saturated steam at 250 lb. per sq. in. abs.

$$\text{Total heat (from Table)} = 209{\cdot}1 + 463{\cdot}1 = 672{\cdot}2 \text{ C.H.U.}$$
$$\text{Pressure} = 250 \times 144 \text{ lb. per sq. in.}$$
$$\text{Volume per pound} = 1{\cdot}85 \text{ cu. ft.}$$
$$\therefore \text{ Heat to external work} = \frac{250 \times 144 \times 1{\cdot}85}{1400}$$
$$= 47{\cdot}5 \text{ C.H.U.}$$
$$\therefore \text{ Internal energy} = 672{\cdot}2 - 47{\cdot}5$$
$$= 624{\cdot}7 \text{ C.H.U.}$$

Example 2.—Find the internal energy of 1 lb. of wet steam, dryness 0·9, at 250 lb. per sq. in. abs.

$$\text{Total heat} = 209{\cdot}1 + 0{\cdot}9 \times 463{\cdot}1 = 625{\cdot}9 \text{ C.H.U.}$$
$$\text{Volume per pound} = 0{\cdot}9 \times 1{\cdot}85 = 1{\cdot}66 \text{ cu. ft.}$$
$$\text{Heat to external work} = \frac{250 \times 144 \times 1{\cdot}66}{1400}$$
$$= 42{\cdot}7 \text{ C.H.U.}$$
$$\therefore \text{ Internal energy} = 625{\cdot}9 - 42{\cdot}7$$
$$= 583{\cdot}2 \text{ C.H.U.}$$

From the following table, p. 37, it will be seen that saturated steam under a given pressure has a fixed temperature, also that the temperature and density increase with the pressure. But it will be further noticed that the total heat increases in a very slow ratio compared with the pressure and temperature, there being only a very small increase of total heat per pound of steam as the pressure increases. This is an important point in practice when considered in reference to coal consumption, for it shows that it is not much more costly in fuel to generate high-pressure steam than low-pressure steam, weight for weight; but we shall see further on that far more work can be obtained from high-pressure steam when used expansively than from the same weight of low-pressure steam, and hence the economy of high-pressure steam.

Drawbacks to the use of high-pressure steam are extra

Absolute Pressure lb. per sq. in.
0·5
1
2
3
4
5
6
7
8
9
10
14·69
15
20
25
30
35
40
50
60
70
80
90
100
110
120
130
140
150
160
180
200
220
240
250
260
280
300
350
400
500

TABLE III.—PROPERTIES OF SATURATED STEAM

Centigrade Units			Fahrenheit Units			Volume of 1 lb (cu. ft.)
Saturation Temp.	Sensible Heat	Latent Heat	Saturation Temp.	Sensible Heat	Latent Heat	
26·41	26·34	580·4	79·5	47·4	1045	640·5
38·74	38·63	573·8	101·7	69·5	1033	333·1
52·27	52·16	566·5	126·1	93·9	1020	173·5
60·83	60·7	561·8	141·5	109·3	1011	118·6
67·23	67·1	558·3	153·0	120·8	1005	90·54
72·38	72·26	555·4	162·3	130·0	999·8	73·44
76·72	76·61	552·9	170·1	137·9	995·3	61·91
80·49	80·39	550·8	176·9	144·7	991·4	53·59
83·84	83·75	548·8	182·9	150·8	987·8	47·3
86·84	86·76	547·1	188·3	156·3	984·6	42·36
89·58	89·51	545·5	193·3	161·1	981·9	38·39
100·0	100·0	539·3	212·0	180·0	970·7	26·79
100·6	100·6	538·9	213·1	181·0	970·2	26·27
108·9	108·9	533·9	228·0	196·1	961·0	20·08
115·6	115·8	529·6	240·1	208·4	953·3	16·29
121·3	121·5	526·0	250·3	218·8	946·7	13·72
126·3	126·6	522·8	259·3	227·8	941·0	11·86
130·7	131·1	519·9	267·2	235·9	935·6	10·48
138·3	138·9	514·7	280·9	250·0	926·3	8·50
144·8	145·5	510·2	292·6	261·9	918·8	7·162
150·5	151·4	506·2	302·8	272·4	910·9	6·196
155·5	156·6	502·6	311·9	281·9	904·2	5·466
160·1	161·4	499·2	320·2	290·5	898·0	4·891
164·3	165·7	496·1	327·7	298·5	892·2	4·429
168·2	169·7	493·2	334·7	305·7	886·9	4·043
171·8	173·5	490·4	341·2	312·5	881·8	3·727
175·1	177·1	487·8	347·2	318·7	877·0	3·455
178·3	180·4	485·3	353·0	324·9	872·3	3·221
181·3	183·6	482·9	358·4	330·6	867·9	3·016
184·2	186·6	480·6	363·5	335·9	863·7	2·836
189·5	192·3	476·3	373·1	346·1	855·6	2·536
194·4	197·5	472·2	381·8	355·5	848·0	2·293
198·9	202·3	468·4	390·0	364·2	840·9	2·093
203·1	206·9	464·8	397·4	372·4	834·0	1·926
205·1	209·1	463·1	401·0	376·3	830·7	1·852
207·0	211·2	461·3	404·5	380·1	827·4	1·783
210·8	215·3	458·0	411·1	387·3	821·2	1·660
214·3	219·1	454·9	417·4	394·2	815·2	1·552
222·5	228·1	447·4	431·8	410·1	801·0	1·336
229·8	236·2	440·6	444·7	424·6	787·5	1·172
242·6	250·7	428·3	467·1	450·1	763·1	0·940

cost of boiler and steam piping, more trouble with pipe joints, and a more costly engine. On the other hand, owing to the reduced steam consumption the boiler, piping, engine, and condensing plant will be smaller than for low-pressure steam. In the case of a large installation there may also be an important saving in total space occupied.

Example.—A cylinder contains 15 cu. ft. of steam at 40 lb. absolute pressure : find the weight of this volume of the steam.

By Table III steam at 40 lb. absolute pressure occupies 10·5 cu. ft. per lb.

Then, 10·5 cu. ft. of steam at 40 lb. pressure weigh 1 lb.

1 „ „ „ „ $\frac{1}{10 \cdot 5}$ lb.

15 „ „ „ „ $15 \times \frac{1}{10 \cdot 5}$ lb.

$= 1 \cdot 428$ lb.

Temperature of Mixtures—Condensing Water

Example 1.—If 1 lb. of water at 212° F. be mixed with 5 lb. of water at 50° F., find the temperature of the mixture.

NOTE.—In order to avoid confusion in problems of this kind, it is necessary to remember that the *total heat* in water or steam is always reckoned from 32° F. or 0° C. Hence it is necessary to subtract 32 from the temperature given in Fahrenheit degrees.

Let t = temperature required. Then

Total heat in 1 lb. of water at 210° + Total heat in 5 lb. of water at 50° = Total heat in 6 lb. of water at t°

$$1(212-32)+5(50-32)=6(t-32)$$
$$180+90=6t-192$$
$$6t=462$$
$$t=77° \text{ F.}$$

Example 2.—How much water at 55° F. must be mixed with 1 lb. of *water* at 212° F. so that the resulting temperature of the mixture may be 105° F. ?

Let W = weight of water required ; then

Total heat in 1 lb. of water at 212° + Total heat in W lb. of water at 55° = Total heat in (W+1) lb. of the mixture at 105°

$$1(212-32)+W(55-32)=(W+1)(105-32)$$
$$180+23W=73W+73$$
$$50W=107$$
$$W=2 \cdot 14 \text{ lb.}$$

In this connection it is interesting and important to compare the difference in the weight of water required to cool a given weight of *water*, with that required to cool the same weight of *steam* at the same temperature.

In the following example it is shown that it takes ten times as much water to cool 1 lb. of steam at 212° as it takes to cool the same weight of water at 212° to the same final temperature of 105°.

Example 3.—How much water at 55° F. will be necessary to condense 1 lb. of *steam* at 212° so that the resulting temperature in the vessel shall be 105° F., assuming condensation takes place at the pressure due to the temperature of the steam ?

Let W = weight of water required ; then

$$\begin{array}{c}\text{Total heat of 1 lb.}\\ \text{of steam at 212°}\end{array} + \begin{array}{c}\text{Total heat in W lb.}\\ \text{of water at 55°}\end{array} = \begin{array}{c}\text{Total heat in (W+1) lb.}\\ \text{of water at 105°}\end{array}$$

$$1151 + W(55-32) = (W+1)(105-32)$$
$$1151 + 23W = 73W + 73$$
$$50W = 1078$$
$$W = 21{\cdot}56 \text{ lb.}$$

Compare this answer with that in Ex. 2 above.

Example 4.—Find the temperature of the mixture when 21·5 lb. of condensing water at 55° F. are used per lb. of steam at atmospheric pressure.

Let t = the temperature required ; then

$$\begin{array}{c}\text{Total heat in 1 lb.}\\ \text{of steam at 212°}\end{array} + \begin{array}{c}\text{Total heat in 21·5 lb. of}\\ \text{condensing water at 55°}\end{array} = \begin{array}{c}\text{Total heat in 22·5 lb.}\\ \text{of mixture}\end{array}$$

$$1151 + 21{\cdot}5(55-32) = 22{\cdot}5(t-32)$$
$$1151 + 494{\cdot}5 = 22{\cdot}5t - 720$$
$$22{\cdot}5t = 2365{\cdot}5$$
$$t = 105{\cdot}1° \text{ F.}$$

Example 5.—Find the weight of cooling water required to condense 1 lb. of steam at 1 lb. per sq. in. abs., dryness 0·86, if the final temperature of the condensed steam is 92° F. and the rise of temperature of the cooling water is 30° F.

Total heat of 1 lb. of steam (from 32° F.) $= 69{\cdot}5 + 0{\cdot}86 \times 1033$
$= 958{\cdot}5$ B.Th.U.

This is the heat lost if the steam is cooled to 32° F.

∴ Actual heat lost by each pound of steam $= 958{\cdot}5 - (92-32) =$ 898·5 B.Th.U.

Let w = weight of cooling water required (pound).

Heat gained by cooling water = heat lost by steam

$$w \times 30 = 898{\cdot}5$$
$$w = 29{\cdot}95 \text{ lb.}$$

Example 6.—To what temperature can 40 lb. of water be raised by the condensation of 1 lb. of steam at 150 lb. per sq. in. abs., superheated by 60° F., if the condensed steam leaves at 120° F. and the initial temperature of the water is 65° F. ? Specific heat of the superheated steam$=0{\cdot}52$.

Total heat of 1 lb. of dry saturated steam$=330{\cdot}6+867{\cdot}9=1198{\cdot}5$ B.Th.U.

Extra Heat required for superheating$=0{\cdot}52\times 60=31{\cdot}2$ B.Th.U.

$\therefore$ Total heat of 1 lb. of superheated steam$=1229{\cdot}7$ B.Th.U.

$\therefore$ Heat lost per pound of steam$=1229{\cdot}7-(120-32)=1142$ B.Th.U.

Let t=final temperature of water

$$40(t-65)=1142$$
$$t=93{\cdot}6° \text{ F.}$$

CHAPTER V

EXPANSIVE WORKING

When steam engines are required to exert their full power for a short period—as happens, for example, with the locomotive in mounting an incline—steam is admitted to the cylinder at full pressure through the greater part of the stroke, without regard to economy in the consumption of steam or fuel. But this is not the way in which steam is used for any length of time in well-constructed and well-managed engines ; and although extra work is obtained from the engine by neglecting to use the steam expansively, it is being very dearly paid for in the excessive proportion of steam and fuel consumed compared with the extra work done, as we shall now proceed to show.

The Expansion Curve.—When the supply of steam to the cylinder of a steam engine is cut off before the stroke is completed the steam expands, the pressure falling as the volume increases. Measurements from actual indicator diagrams show that in most cases the expansion *nearly* follows the law 'pressure varies inversely as the volume,' and for purposes of preliminary calculation this is assumed to be the law of expansion. Thus an increase of volume from 2 cu. ft. to 3 cu. ft. would reduce the pressure to two-thirds. The expansion curve thus obtained is known as a hyperbola and the expansion is known as 'hyperbolic' expansion. (Boyle's Law, dealt with later, refers to gases only and steam should never be referred to as expanding 'according to Boyle's Law,' as the temperature falls continuously during expansion.)

Work done by Steam used expansively.—We have seen that the work done per pound of steam without expansion at high pressures only slightly exceeds that done by the same *weight* of steam at low pressures. We will now call

attention to the increased work which may be obtained from high-pressure steam when advantage is taken of its expansive properties.

Let 1 lb. of steam at 100 lb. per sq. in. abs. be admitted to a cylinder (Fig. 23), when the piston P is at the end *a* of the cylinder, and let the supply of steam be continued for one-fourth of the stroke, namely, till the piston reaches *b*, when we will suppose it just contains 1 lb. of steam. The supply is now cut off and the piston is driven for the remainder of the stroke by the expansive force of the steam thus enclosed. At the end of the stroke the steam occupies four times its original volume, and its pressure is now one-fourth its original or initial pressure, and the work done by the 1 lb. of steam will be clearly shown by the aid of a diagram. Thus let *ap* (Fig. 24) be drawn to any convenient scale of pressures to equal 100 lb.; and make av_1 equal to 4·43 to any other convenient scale. (NOTE.—1 lb. of steam at 100 lb. pressure absolute occupies 4·43 cu. ft., and if we assume the area of the piston=1 sq. ft., then length av_1 =4·43 ft.) Produce the line to av_4, making av_4=4 times av_1. Complete the figure by the graphical method (p. 45).

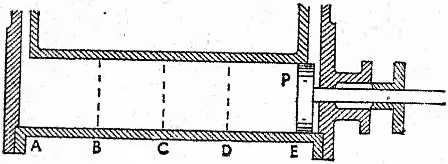

FIG. 23.

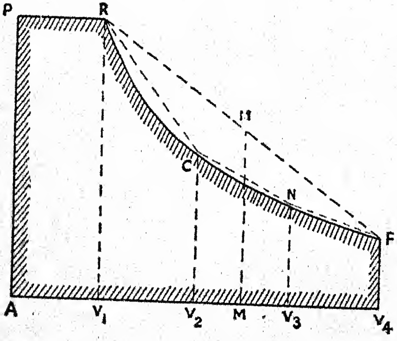

FIG. 24.

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Now the whole work done by the steam is equal to the area of the figure pav_4fr, and this whole area is made up of two parts, namely :

(1) area pav_1r=work done during admission ;
(2) area rv_1v_4f=work done during expansion.

Hence, by making use of the expansive properties of steam, we obtain the additional work out of it represented by the latter area.

To find the area of the figure would be a simple process if the line rf (Fig. 24) had been a straight line instead of a curve, for then the area of the admission portion$=ap \times av_1$; and the area of the expansion portion$=v_1v_4$ multiplied by the mean height mm. But the curve falls below this line, hence the area thus obtained is too large, and the greater the expansion the greater the error. Much greater accuracy, however, is secured by this method if several divisions are taken, as shown by the dotted lines rc, cn, nf, in Fig. 24, and the greater the number of divisions taken the greater the accuracy of the result. This is practically the method used by engineers in finding the area of the indicator diagram, the figure being divided into ten equal portions, as explained below.

The work done in the cylinder during expansion is calculated from the *average* or mean force acting on the piston during the stroke. This is obtained by finding the mean pressure in pounds per square inch acting during the stroke and multiplying this by the area of the piston in square inches.

The method of drawing a diagram showing the variation of pressure during the stroke is illustrated by the following example :

Example.—Steam at 85 lb. per sq. in. boiler pressure, or 100 lb. pressure per square inch absolute, is admitted to a cylinder 5 ft. long, and cut off at $\frac{1}{5}$ of the stroke.

Let OM=line of pressures, and on it mark a scale of pressures, say $\frac{1}{10}$ in.=5 lb. Let ON=line of volumes to scale of, say, $\frac{3}{8}$ in.=1 ft. of stroke of piston, and divide this line into ten equal parts. Complete the rectangle OMaA. Then OA=the volume of the steam and

Aa the pressure, at the point where the steam is cut off. To find the pressures Bb, Cc, etc., at B, C, D, etc., corresponding to the successive volumes OB, OC, OD, etc., advancing by distances along ON of 0·5 ft. Since the pressure at any point is inversely as the volume :

$$\text{Pressure at B}=\frac{OA}{OB}\times\text{initial pressure}=\frac{2}{3}\times 100=66{\cdot}66$$

$$\text{,, }\quad C=\frac{OA}{OC}\quad\text{,, ,,}\quad=\frac{2}{4}\times 100=50{\cdot}00$$

$$\text{,, }\quad D=\frac{OA}{OD}\quad\text{,, ,,}\quad=\frac{2}{5}\times 100=40{\cdot}00$$

$$\text{,, }\quad E=\frac{OA}{OE}\quad\text{,, ,,}\quad=\frac{2}{6}\times 100=33{\cdot}33$$

$$\text{,, }\quad F=\frac{OA}{OF}\quad\text{,, ,,}\quad=\frac{2}{7}\times 100=28{\cdot}57$$

$$\text{,, }\quad G=\frac{OA}{OG}\quad\text{,, ,,}\quad=\frac{2}{8}\times 100=25{\cdot}00$$

$$\text{,, }\quad H=\frac{OA}{OH}\quad\text{,, ,,}\quad=\frac{2}{9}\times 100=22{\cdot}22$$

$$\text{,, }\quad N=\frac{OA}{ON}\quad\text{,, ,,}\quad=\frac{2}{10}\times 100=20{\cdot}00$$

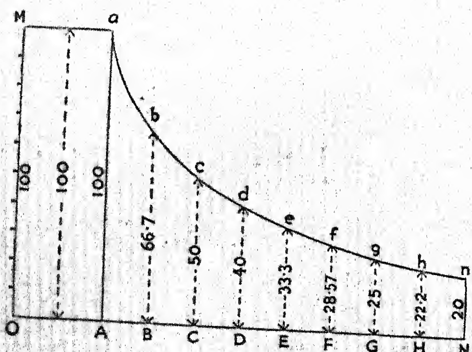


FIG. 25.

The above method of finding the pressure at any point during the expansion when the initial pressure is given may be expressed as follows : Multiply the initial pressure in pounds per square inch by the length of stroke to point of

cut-off, and divide by the distance of the given point from the beginning of the stroke.

The hyperbolic curve may be described without any calculation by the following simple geometrical method.

Draw the lines OM and ON as before. (NOTE.—The point O in the line OM is the zero of pressure, and not the point through which the line of atmospheric pressure passes.) Complete the parallelogram OM*a*A as in Fig. 27. Produce M*a* parallel to ON. To find the pressure at any point B corresponding to the volume OB, draw the vertical

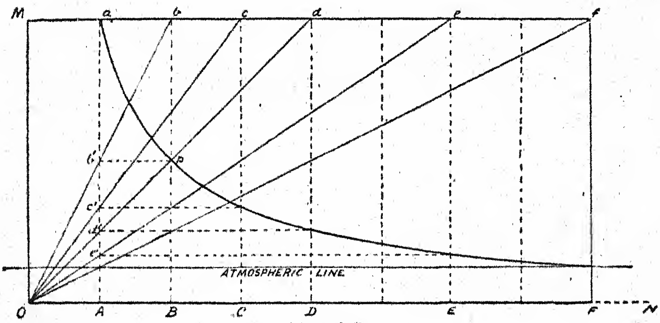


FIG. 26.

B*b* and join O*b*, cutting A*a* in *b′*. Then the horizontal through *b′* to cut the vertical B*b* gives a point *p* in the curve. Any number of other points may be obtained in the same way, and the curve drawn through the points may be completed.

Having drawn the diagram, the mean pressure in pounds per square inch can be obtained by using mid-ordinates, as shown in Fig. 30. This mean pressure can also be calculated from the formula

$$\text{Mean pressure} = P \times \frac{1 + \text{hyp. log. } R}{R}$$

where P = initial pressure of steam (pounds per square inch absolute)

R = ratio of expansion.

The hyperbolic logarithm of R is obtained by multiplying the common logarithm by 2·3 or from a table of hyperbolic logarithms.

The *ratio of expansion* is the ratio of the final volume to the volume at cut-off, e.g. if cut-off is at 0·2 of the stroke, R=5.

To find the *effective* work done during the stroke we must allow for the fact that the pressure on the other side of the piston is not zero. The *mean effective pressure* is found by subtracting this pressure (known as the *back pressure*) from the mean forward pressure. Thus

$$\text{Mean effective pressure } P\times\frac{1+\text{hyp. log. R}}{\text{R}}-p_b$$

where p_b is the back pressure (pounds per square inch absolute).

It will be seen that with a given initial pressure the area of the diagram, and hence the mean effective pressure, depends upon the ratio of expansion and the back pressure. Increasing the ratio of expansion, other things remaining the same, decreases the mean effective pressure, so that a larger cylinder is required for the same work per stroke. Thus one disadvantage of expanding the steam is that the engine becomes more costly. Against this we must set the fact that less steam is used per stroke, and just as we judge the value of commodities by their price per pound or per gallon, so we must judge the value of expansion by the work done *per cubic foot of steam used*, or if we are comparing engines using steam of different pressures we must judge by the work done *per pound of steam used*, since the total heat per pound does not vary much with pressure.

Now we are going to assume at first that a diagram such as Fig. 25 represents the actual variation of pressure in the cylinder, i.e. that the pressure is constant up to cut-off, the steam expands according to the law *p.v.=constant*, exhaust takes place at the end of the stroke, and that there is no clearance volume and no compression near the end of the exhaust stroke. Since we make these assumptions, or

hypotheses, such a diagram is known as a *hypothetical indicator diagram.*

Let us take a cylinder with an area of 1 sq. ft. and a stroke of 1 ft. Assume that the initial pressure is 100 lb. per sq. in. abs. and the back pressure is 20 lb. per sq. in. abs. Using the formula for mean effective pressure, we can tabulate results as follows :

Fraction of Stroke at cut-off	Ratio of Expansion	Mean Effective Pressure, lb. per sq. in.	Mean Driving Force on Piston, lb.	Work done per stroke, ft.-lb.	Vol. of Steam used, cu. ft.	Work done per cu. ft. of steam used, ft.-lb.
0·1	10	13	1,870	1,870	0·1	18,700
0·2	5	32·2	4,640	4,640	0·2	23,200
0·4	2·5	56·8	8,170	8,170	0·4	20,420
0·6	1·67	70·4	10,130	10,130	0·6	16,900
0·8	1·25	77·6	11,190	11,190	0·8	13,990

We see from the last column of the table that as we increase the ratio of expansion from 1·25 to 5 the work done per cubic foot of steam increases continuously, but if the ratio is increased from 5 to 10 instead of more we get less work per cubic foot of steam in this case. Reference to Fig. 26 will explain this, as if the steam is cut off at 0·1 of the stroke the final pressure will be $100 \div 10 = 10$ lb. per sq. in., which is less than the back pressure. Hence in an engine cylinder the steam should never be cut off so early as to expand to a pressure below that of the back pressure acting against the piston. This means that the ratio of expansion should never be greater than the initial pressure divided by the back pressure (5 in the above case). Owing to the effects of friction, condensation of steam in the cylinder, and cost of engine, the ratio adopted in practice is less than this, but for steam turbines, where the losses are much smaller, the actual ratio of expansion is very nearly the same as the calculated ratio.

Now let us take the same cylinder as above, and compare the work done per *pound* of steam with a cut-off of 0·2

with pressures of 100 lb. per sq. in. and 200 lb. per sq. in. respectively.

Referring to Table III, the volume of 1 lb. of steam at 100 lb. per sq. in. is 4·43 cu. ft. and at 200 lb. per sq. in. is 2·29 cu. ft. Working as above, the work done per cubic foot at 100 lb. per sq. in. is 23,200 ft.-lb. and the work done *per pound* of steam at 100 lb. per sq. in. is $23{,}200 \times 4{\cdot}43 = 102{,}600$ ft.-lb. Similarly, the work done per cubic foot at 200 lb. per sq. in. is 60,800 ft.-lb. and the work done *per pound* of steam at 200 lb. per sq. in. is $60{,}800 \times 2{\cdot}29 = 139{,}000$ ft.-lb. (37 per cent. increase). The difference is still more pronounced when the back pressure is reduced, as in a condensing engine.

We have assumed the same ratio of expansion in each case, but with the higher pressure a larger ratio of expansion can be used with advantage, with a still greater gain in economy.

Obviously, more work per pound of steam means less steam for a given amount of work and there will be a saving in fuel consumption corresponding with the saving in weight of steam. In practice it would be necessary to set against the above result :

(1) Probable increased initial condensation of steam in the cylinder, with the higher pressure and greater expansion.

(2) Increased initial stresses on the engine.

Back Pressure.—Back pressure has a considerable influence on the total work done by a given weight of steam.

Suppose the piston of a steam engine to be acted upon on one side by steam of 45 lb. pressure absolute, and, if it be possible, let there be no pressure at all acting on the other side. Then, if the pressure of the steam were maintained uniform throughout the stroke, the diagram of pressures and volumes, or, in other words, the diagram of work, would be a simple rectangle, thus (Fig. 27) :

But in ordinary steam engines without a condenser, as the locomotive and many small factory engines, when the steam acts on one side of the piston, communication is open with the atmosphere through the exhaust passage on

the other side, and it is therefore exposed to a back pressure of 15 lb. per sq. in. (Fig. 28). The effective pressure is therefore 45—15=30 lb. per sq. in. ; and the effect on the diagram is to remove all the lower part from zero to 15 lb., and thus reduce the area of the figure, and therefore also the effective work done. In practice there is an additional back pressure of 2 to 4 lb., due to incompleteness of exhaust, making a total back pressure of 17 lb. to 19 lb. per sq. in. It may be much more than this with high-piston speeds. If, however, the cylinder were put, during exhaust, into communication with a condenser, then a large portion of the atmospheric pressure is removed, and a back pressure of

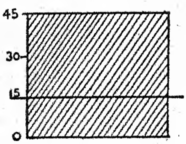


FIG. 27.

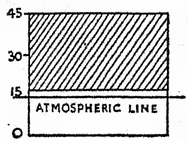


FIG. 28.

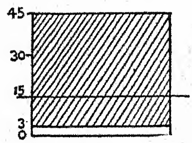


FIG. 29.

not more than about 3 lb. abs. will now oppose the motion of the piston. In this case the area of the figure representing the effective work done will be extended down to within about 3 lb. of the zero line (Fig. 29) ; the gain of work being proportional to the gain of area ; while the weight of steam used in each case is clearly the same.

Effective pressure=Difference between pressures on each side of piston.

Mean Pressure.—To find the mean effective pressure of steam per square inch on the piston, by measurement from the indicator diagram :

(1) Divide the line of volumes into ten equal parts.

(2) Measure the height of the figure at the centre of each division by the scale of pressures.

(3) Add the measurements together, and divide the sum by ten. The result gives the mean effective pressure per square inch on the piston.

To find the *total* mean pressure on the piston, multiply

the mean pressure per square inch by the area of the piston in square inches.

Then the mean pressure on the piston in pounds, multiplied by the length of stroke in feet, gives the area of the figure, or the work done per stroke in foot-pounds.

Example.—Find the mean effective pressure in the cylinder of a condensing steam engine when the pressure of steam on admission is 80 lb. abs., cut off at one-fourth of the stroke. Back pressure 3 lb. per sq. in.

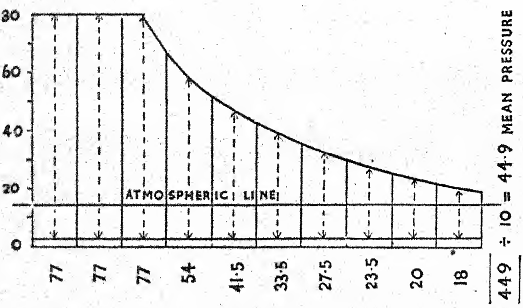


FIG. 30.

The same result might have been obtained for the theoretical diagram by using the formula given on p. 45.

The following is useful for reference in obtaining the hypothetical mean pressure :

IV. TABLE OF MEAN PRESSURES

Number of times steam is expanded $=\frac{\text{final volume}}{\text{initial volume}}$	Mean pressure throughout stroke. Initial pressure = 1	Number of times steam is expanded $=\frac{\text{final volume}}{\text{initial volume}}$	Mean pressure throughout stroke. Initial pressure = 1
$1\frac{1}{3}$	0·964	6	0·465
$1\frac{1}{2}$	0·937	7	0·421
2	0·846	8	0·385
3	0·699	9	0·355
4	0·596	10	0·330
5	0·522		

Subtract the back pressure from result obtained by above table, and the remainder will give the mean *effective* pressure.

Example.—Steam at 100 lb. abs. is expanded down to 20 lb., back pressure 17 lb.; find the mean effective pressure.

Here $\frac{100}{20}=5=$number of expansions

mean pressure (by table)$=0{\cdot}522$

$100\times0{\cdot}522=52{\cdot}2$ lb. mean absolute pressure, or
$52{\cdot}2-17=35{\cdot}2$ lb. per sq. in. mean effective pressure.

The actual indicator diagram obtained from a steam engine differs from the hypothetical diagram in a number of ways, depending upon the type of engine and the method by which it is actuated.

(*a*) The admission line slopes somewhat, the amount of slope being greater for high speeds of revolution than for slow speeds. The steam has to enter through an opening of varying size, and in order that it may enter the cylinder the pressure must be less than that in the steam chest. As the piston increases its speed the steam must flow faster through the opening, and the drop of pressure necessary to cause it to flow increases. This drop of pressure is known as 'wiredrawing.' The term has been in use from the early days of the steam engine and is still used, although it does not define what is actually happening.

(*b*) The exhaust valve opens before the end of the stroke in order that the pressure in the cylinder may drop to the back pressure before the exhaust stroke commences.

(*c*) In order to bring the reciprocating parts to rest at the end of the stroke without shock, the exhaust valve closes before the end of the exhaust stroke, trapping some of the steam, which acts in a somewhat similar manner to a buffer spring. High-speed steam engines require a higher degree of compression than slow-speed engines, since the inertia force due to the reciprocating parts is proportional to the square of the speed.

Two diagrams are shown (Fig. 31) from a double-acting vertical engine, one from the top end and one from the bottom end. The mean pressure in the cylinder is taken as the average of these diagrams. When both diagrams are taken by the same indicator, and on the same card, they appear on the card as shown in the figure.

The mean pressure is obtained by drawing lines perpendicular to the atmospheric line through the ends of the diagrams at A and B, and dividing the distance AB into ten equal parts. The mean pressure is then obtained as in the previous example.

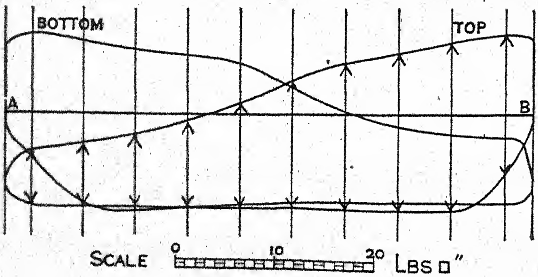


FIG. 31.

Fig. 31 shows the method of measuring the pressures for the diagram from the top end of the cylinder.

$$\text{Mean pressure of top diagram} = 10{\cdot}85$$
$$\text{,, ,, bottom diagram} = 11{\cdot}18$$

$$\text{Average mean pressure} = \frac{22{\cdot}03}{2} = 11{\cdot}0$$

The ratio of the actual mean effective pressure to the hypothetical mean effective pressure is known as the 'diagram factor,' which is, of course, always less than unity. The values of the diagram factors for different types of engine are known from experience, and enable the designer to predict the actual horsepower of an engine with a fair degree of accuracy.

Indicated Horsepower.—The diagram representing the work done on the piston has been called an indicator diagram. From this diagram, having obtained the mean pressure, the work done per stroke may be found. The work done per minute=the work done per stroke×number of working strokes per minute. The result may be expressed in horsepower by dividing the work done per

minute by 33,000. The horsepower obtained from the indicator diagram in this way is called the *Indicated Horsepower.* In the case of a steam engine each stroke is a working stroke, and the number of working strokes per minute is therefore twice the number of revolutions per minute. In a four-stroke gas engine (see Chapter XX) there is only one working stroke in every two revolutions (assuming no missed explosions) and consequently the number of working strokes per minute is half the number of revolutions per minute.

Thus, I.H.P. = mean effective pressure (lb. per sq. in.) $\times$ area of piston (sq in.) $\times$ stroke (ft.) $\times$ number of working strokes per minute $\div$ 33,000.

It is important that the correct units should be used when making a calculation from this equation.

It will be seen from the above equation that we can reduce the size of the steam engine required for a given horsepower: (*a*) by increasing the mean effective pressure by the use of high-pressure steam and low back pressure, (*b*) by increasing the revolutions per minute. The early engines using low-pressure steam and running at a comparatively small number of revolutions assumed the type of the massive beam engine. The modern engine develops the same power with high pressures and high piston speeds, and its dimensions are therefore proportionally decreased.

The horsepower of a compound engine is obtained in practice by finding the horsepower exerted in each cylinder separately from the indicator diagrams by the method above explained, and adding the results together; the sum then gives the total indicated horsepower.

Example 1.—Find the indicated horsepower of an engine with a cylinder 12 in. diameter, length of stroke 18 in., number of revolutions per minute 90, mean effective pressure per square inch on piston 40 lb.

Mean effective pressure = 40 lb. per sq. in.

Area of piston $= 0{\cdot}785 \times 144 = 113$ sq. in.

Stroke $= 1{\cdot}5$ ft.

Number of working strokes per minute $= 2 \times 90 = 180$

$$\therefore \text{I.H.P.} = \frac{40 \times 113 \times 1{\cdot}5 \times 180}{33\,000} = 37$$

Example 2.—An engine is required to indicate 37 horsepower with a mean effective pressure on piston of 40 lb. per sq. in., length of stroke 18 in., number of revolutions per minute 90 ; find the diameter of the cylinder.

Let d=diameter of cylinder (in.)

Area of piston=$0{\cdot}785d^2$ (sq. in.)

Stroke=$1{\cdot}5$ ft.

$$\therefore \frac{40\times 0{\cdot}785d^2\times 1{\cdot}5\times 180}{33{,}000}=37$$

$$d^2=\frac{33{,}000\times 37}{40\times 0{\cdot}785\times 1{\cdot}5\times 180}=144$$

$$d=12 \text{ in.}$$

The indicated horsepower is obtained from the work done *in the cylinder.* Some of this work is wasted in friction of piston, stuffing-box, crosshead, and bearings, the remainder being the horsepower which can be delivered by the engine to whatever it is designed to drive. This *useful* horsepower is known as the *brake horsepower,* from the fact that it can be measured by a brake of some kind applied to the engine.

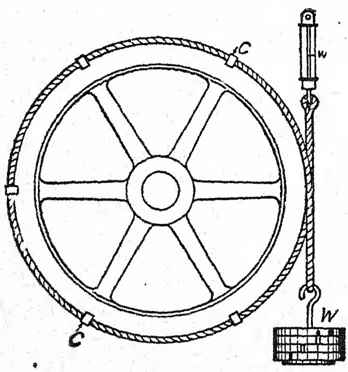


FIG. 32.

The most convenient and easily fitted type of brake is the rope brake, illustrated in Fig. 32, which can be applied to the flywheel of a moderate-sized engine running at speeds up to 300 r.p.m. It consists usually of one or two strands of rope making a complete turn round the flywheel, the upper end being connected to a spring balance and the lower end loaded with weights W. The ropes are held apart by wooden blocks C. When the engine is running steadily at the required load the whole of the power developed is absorbed by friction at the flywheel rim. Since the whole of the power is thus converted into heat,

the rim must be water cooled if more than a small horsepower is being absorbed.

The net resistance at the rim of the flywheel is $W-w$, where w is the reading of the spring balance.

If D is the effective diameter of the brake in feet, and N is the speed in revolutions per minute, then

$$\text{B.H.P.}=\frac{\pi DN(W-w)}{33{,}000}$$

The *mechanical efficiency* of an engine is the ratio of work obtained from the crankshaft to the work done in the cylinder, and is expressed as follows:

$$\text{Mechanical efficiency}=\frac{\text{Brake horsepower}}{\text{Indicated horsepower}}$$

It is a measure of the *mechanical* perfection of the engine. The greater the friction of the mechanism is the less will be the mechanical efficiency.

Example.—The I.H.P. of a gas engine is 280, and the B.H.P. is 210. What is the mechanical efficiency ?

$$\text{Mechanical efficiency per cent. is }\frac{210\times 100}{280}=75\text{ per cent.}$$

Clearance in the Cylinder.—When the piston in a cylinder is at the end of its stroke it does not *touch* the end or cover of the cylinder, but there is always a certain space left between them to prevent the danger of their coming into actual contact. In addition to this is the passage between the face of the slide valve and the cylinder by which the steam is conducted to the cylinder. These two spaces (marked *cc*, Fig. 33), which make up the whole space between the face of the valve and the face of the piston when the piston is at the end of its stroke, are called the *clearance volume*.

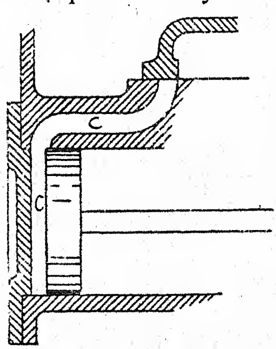


FIG. 33.

Let the volume displaced by the piston during its stroke $=9$ cu. ft.; and the volume of the clearance $=1$ cu. ft. Then, when steam is admitted, 1 cu. ft. is used to fill the clearance space before the piston moves; and if steam is used at full pressure throughout the stroke, 9 cu. ft. more is required to displace the piston. Thus 1 cu. ft. in every 10 cu. ft. of steam passes through the engine without doing any work, representing a loss of 10 per cent.

But suppose the steam is cut off at $\frac{1}{9}$th of the stroke. Then, during admission there is first 1 cu. ft. of steam to fill the clearance, and which so far does no work on the piston, and then 1 cu. ft. to displace the piston, when the steam is cut off. There are now 2 cu. ft. of steam at initial pressure enclosed in the cylinder. Expansion commences, and at the end of the stroke the volume occupied by the steam will evidently be 10 cu. ft. Hence, pressure of steam at end of stroke $=\frac{2}{10}$ths, or $\frac{1}{5}$th of initial pressure.

If there had been *no* clearance, then we should have had 1 cu. ft. of steam in the cylinder at point of cut-off, which would expand to 9 cu. ft. with a terminal pressure of $\frac{1}{9}$th, the initial pressure.

Suppose the initial pressure had been 180 lb. absolute. Then, neglecting the effect of clearance, the terminal pressure $=\frac{180}{9}=20$ lb. per sq. in. abs.

Including the effect of clearance, the terminal pressure $=\frac{180}{5}=36$ lb. per sq. in. abs., or nearly twice the terminal pressure obtained neglecting clearance.

Although the steam required to fill the clearance space does no work on the piston during admission, yet when cut-off takes place the piston receives the advantage of the expansive force of this steam, and its effect in increasing the total work done is shown by the shaded part of the diagram (Fig. 34).

To draw the diagram, set off ac = the stroke of the piston, to any scale and divide it into nine equal parts, construct the curve p, 20, by the graphical method from the point a, representing the expansion of steam of volume ab and pressure bp. To the left of a draw ad, making $ad=\frac{1}{9}$th ac,

that being the proportion of the volume of the cylinder occupied by the clearance space. Draw the curve *p*, 36,

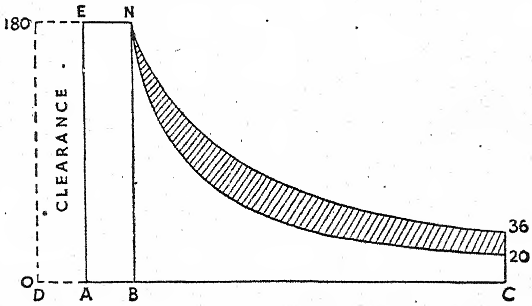


FIG. 34.

from the point *d*, representing the expansion of steam of volume *db* and pressure *bp*.

To draw the indicator diagram, allowing for clearance.— Let AB represent the volume of the cylinder to any scale. Make OA=clearance, say, 5 per cent. of AB. OP represents the *absolute* initial pressure of the steam. Assuming cut-off takes place at one-quarter of the stroke, make AC $=\frac{1}{4}$AB. Draw inclined lines from O, cutting CD and PE, as in Fig. 35, and so obtain points on the expansion curve. Draw FG the back pressure line by making AF=the absolute back pressure. Then the indicator diagram is the figure HDKGF.

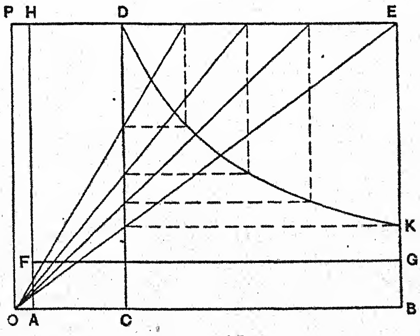


FIG. 35.

The loss by clearance is greater with a very early cut-off.

With 10 per cent. clearance and cut-off at 10 per cent. of the stroke the amount of steam used is increased by 100 per cent., while the work done is only increased by about 30 per cent. If the cut-off is at half stroke the steam used is increased by 20 per cent. and the work done is increased by about 6 per cent. It is sometimes argued that the loss by clearance can be reduced by closing the exhaust valve early and compressing the exhaust steam up to the initial pressure, so that the clearance space is filled with steam at high pressure when admission begins. This ignores the fact that there is a loss of diagram area due to compression, which reduces the net work done. As a matter of fact, both theory and experiment show that with more than a very moderate amount of compression the loss of diagram area more than balances the amount of steam saved.

Cylinder Condensation.—A property of saturated steam (i.e. steam which is not superheated) is that its temperature cannot be reduced without reducing its pressure. If saturated steam is brought into contact with a surface colder than the temperature of the steam condensation begins immediately and continues until the steam in condensing has given up sufficient latent heat to warm the surface up to the temperature corresponding to the pressure of the steam.

If now we assume that the surface is covered with a thin film of water at the temperature of the steam (for example, 358° F. at 150 lb. per sq. in. abs.) and the pressure is now lowered by expansion or exhaust (for example, to 50 lb. per sq. in. abs., the saturation temperature for which is 250° F.), then the water at 358° F. will begin to re-evaporate and will continue to evaporate until it has taken up sufficient latent heat from the hot surface to reduce its temperature to 250° F. During exhaust, where the pressure may be very much lower, the cooling effect of re-evaporation will be still more pronounced.

When steam is admitted to a cylinder which is colder than the entering steam, the steam parts with some of its heat to the cylinder walls, a portion of the steam is con-

densed and deposited on the metallic surface, and more steam from the boiler enters the cylinder to take its place, while the temperature of the cylinder rises to that of the steam in contact with it.

If the steam be supplied to the cylinder at the initial pressure and temperature throughout the whole stroke, and the exhaust port be then opened, the steam will escape into the air, and the pressure in the cylinder will fall to that of the atmosphere, or nearly so.

But the water (which exists more or less as a film), being in contact with the metallic walls of the cylinder at the temperature of the initial steam, will evaporate immediately the pressure is reduced by the opening of the exhaust, and become reconverted into steam at the expense of the heat in the walls of the cylinder, thereby cooling them to the temperature of the steam during exhaust.

The steam thus re-evaporated during exhaust not only absorbs heat, which will have to be made up again from the entering steam during the next stroke, but it passes away to the air without doing any useful work ; in fact, it acts rather as back pressure against the piston.

When the piston reaches the end of its stroke, the boiler steam is readmitted into the cylinder and comes in contact with the cooled surface of the cylinder cover, piston, and steam passages, which have been exposed to the temperature of the exhaust steam, and the same process of condensation and re-evaporation will be repeated.

If the cylinder had been in communication with a condenser instead of with the air, the temperature of the cylinder during exhaust would have fallen still lower, namely, to that due to the decreased pressure in the condenser, and the condensation of the initial steam during admission would have been still greater.

If the steam is cut off at an early point in the stroke, condensation occurs, as before, during admission, while the steam is hotter than the cylinder ; but as the expansion proceeds, a portion of the condensed steam is re-evaporated, the re-evaporation increasing as the pressure decreases

towards the end of the stroke. On the opening of the port to exhaust, the pressure is still further reduced, and re-evaporation is completed.

Condensation, then, takes place during the early part of the stroke, while re-evaporation occurs partly towards the end of the stroke and partly during exhaust. The re-evaporation during expansion behind the piston helps the piston, and increases the total work done ; but the steam re-evaporated during exhaust in a single-cylinder engine passes away to waste. The loss due to condensation of steam in the cylinders of all engines varies from 10 to 50 per cent., or more, of the whole steam consumed, the loss becoming greater as the mean temperature of the cylinder walls becomes less than the temperature of the initial steam.

Condensation in the cylinder increases as the degree of expansion increases, because there is a decreasing mean temperature of the walls with a constant initial temperature of the steam.

The economical advantage of using high-pressure steam is due to the power it possesses of doing work by expanding behind the piston after the supply is cut off from the boiler.

But the temperature of saturated steam varies with the pressure, and, therefore, if, in a single cylinder, steam at high pressure and temperature be admitted and expanded to a low pressure and temperature, the greater the degree of expansion the greater the difference in temperature between the initial steam and the mean temperature of the cylinder walls.

Hence there is a limit to the useful expansion of steam in a single cylinder, owing to the excessive condensation in the cylinder, with high degrees of expansion, resulting in increased consumption of fuel instead of a saving.

The secret of economy is to supply the cylinder with *dry steam*, and to maintain it as dry as possible throughout the stroke, and engineers from Watt's day to the present have striven to accomplish this result.

The means adopted to reduce the amount of water of condensation in the cylinder are :

(1) Obtaining the steam from the boiler as dry as possible, and maintaining it in the dry condition by carefully covering the parts traversed by the steam, on its way to the cylinder, with non-conducting material.

(2) Placing a water-separator in the steam pipe just before entering the engine.

The less water there is to re-evaporate during exhaust, the less will be the cooling effect due to this.

(3) *Jacketing the cylinder* with hot steam (an example of jacketed cylinders is given in Figs. 78 and 81). The addition of the steam jacket reduces the amount of condensation in the cylinder. The jacket is the more necessary the greater the degree of expansion in one cylinder, and the slower the piston speed.

It must be remembered that the jacket can only supply heat to the cylinder walls by the condensation of steam in the jacket, and unless less steam is condensed in the jacket than would otherwise be condensed in the cylinder it will be useless. The advantage of the jacket is that the steam condensed in it is drained away and does not cool the cylinder by re-evaporation. If wet steam is supplied to the engine experience shows that a jacket has no useful effects.

(4) *Compounding the cylinders*, that is, adding one or more separate cylinders into which the steam may be successively expanded, and thereby reducing the variation of temperature in each cylinder.

(5) Increasing the rotational speed of the engine. This reduces the time of contact between steam and cylinder walls, but also increases the number of contacts per second. The main advantage here is that the engine is *smaller* for a given horsepower and hence there is less surface exposed to the steam.

(6) Superheating the steam, that is, applying additional heat to the steam on its way between the boiler and the engine.

If the temperature of the steam is raised, say, 150° F. beyond its saturation temperature, it will not condense if brought into contact with a cold surface until it has lost all its superheat. If by that time the surface has been brought up to the saturation temperature of the steam, no condensation will take place. It must, however, be remembered that some of the heat of the fuel has been expended in superheating the steam; as steam cannot be superheated by waste gases, which are too low in temperature. The secret of the advantage gained lies again in the fact that re-evaporation is reduced or prevented, so that the cylinder walls remain hotter and require less heat during admission. Only a moderate amount of superheat is permissible with ordinary slide valves and Corliss valves, mainly owing to distortion effects, but with piston valves and drop valves high superheats are practicable.

CHAPTER VI

THE RECIPROCATING STEAM ENGINE

Engines which exhaust their steam into the air after it has done work in the cylinder are called non-condensing engines. The locomotive belongs to this type.

In condensing engines there is no escape of exhaust steam into the air, the steam being passed instead into a condenser, where it is cooled and condensed by actual contact with a jet of cold water, or by contact with cold pipes through which cold water is flowing.

The essential parts of all ordinary non-condensing engines, whether the engine be a horizontal or vertical one, are practically the same, the difference in appearance among engines by different makers being due for the most part to a difference in shape or arrangement of the essential details.

The following diagrams (Figs. 36 and 37) give a front view and side view of a small vertical non-condensing steam engine, as in use for various kinds of factory and mill work. The pressure of steam used in such engines is about 60 lb. per sq. in. above the atmosphere.

The action of the parts is as follows : The steam is conducted from the boiler by the steam pipe to the slide jacket or chamber in which the slide valve SV works. Here, by a sliding motion of the slide valve on the face of the ports, the steam passages or ports are alternately opened, admitting steam to one side of the piston, and allowing it to escape from the other side into the air ; or, if a condensing engine, into a condenser. (The exact action of the slide valve will be explained more fully later.) The piston is thus made to move from end to end of the cylinder against the resistance due to the load which is communicated through the piston rod.

Attached to the outer end of the piston rod is the cross-head, having a flat base called a slipper, which slides to and fro between guides, and compels the piston rod to

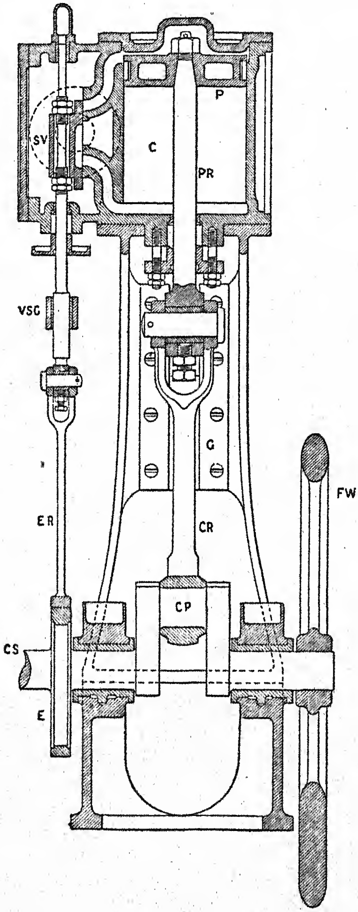


FIG. 36.

C, cylinder ; P, piston ; PR, piston rod ; G, guides ; CR, connecting rod ; CP, c ankpin ; E, eccentric ; ER, eccentric rod ; SV, slide valve ; VSG, valve spindle guide ; CS, crankshaft.

move parallel to the axis of the cylinder, thus preventing the angular action of the connecting rod from bending the piston rod. The connecting rod is attached at one end to

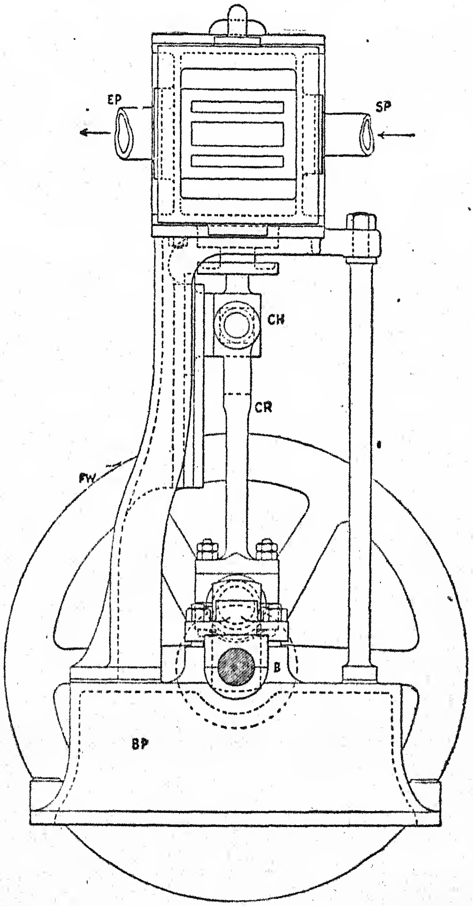


FIG. 37.

CH, crosshead ; CR, connecting rod ; B, bearings ; EP, exhaust pipe ; SP, steam pipe ; BP, bed-plate ; FW, flywheel.

the crosshead by a pin, sometimes called a gudgeon, which passes completely through the block and the fork end of the rod as shown, and at the other end to the crankpin The reciprocating motion of the piston is by this means converted into the circular motion of the crankpin and shaft, and from the shaft by means of a pulley and belt, or by wheel gearing, the power of the engine is transmitted as required. See also Figs. 124 and 125.

Engine Details

The Cylinder.—The cylinder, which is made of cast iron, consists of the cylindrical chamber, bored out perfectly true, and of the slide jacket or valve box. The cylindrical chamber is connected at each end with the slide jacket by passages called *steam ports*, S, through which steam passes to or from the cylinder. The passage between the two steam ports leads to the air, or to a condenser, and is called the *exhaust port*, E. This passage is put in communication with either end of the cylinder as required by means of the slide valve. The ends of the cylinder are closed by covers bolted to the flanged ends. In the example (Fig. 38) the bottom end is cast solid with the body of the cylinder.

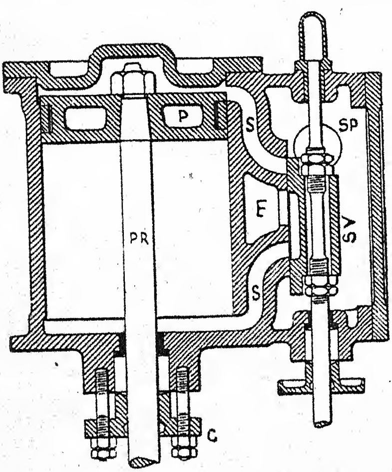


FIG. 38.

SP, steam pipe ; S, steam port ; E, exhaust port ; SV, slide valve ; P, piston ; PR, piston rod ; G, gland.

In order to make the hole in the cover through which the piston rod passes steam-tight, a *stuffing box* is used.

the construction of which will be understood from the figure. The casting is so formed as to leave a small space around the rod, which is filled with some form of flexible material capable of making a steam joint round the piston rod, and the packing is pressed down on the rod by means of a cover or *gland* fitted with two screwed bolts. A similar arrangement of stuffing box and gland is fitted to the slide valve rod ; it is also used for pump rods and other similar purposes.

The steam passages should be made as short as possible, because at each stroke the passage must be filled with its own volume of steam before the steam acts upon the piston. The effect of this has been described under the heading of *Clearance*, on p. 55.

The steam ports must be made large enough to admit sufficient steam to the cylinder during the instant the port is open, otherwise the steam will be *wire-drawn*.

Wiredrawing is the gradual fall of pressure of the steam behind the piston, as it proceeds on its stroke, owing to small and restricted steam passages. Its effect may be illustrated by the diagram (Fig. 39).

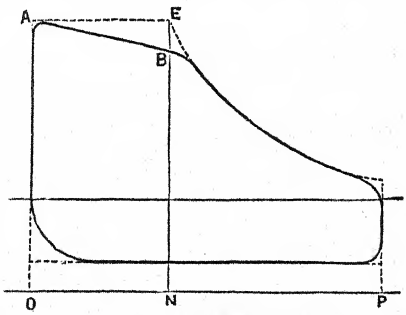


FIG. 39.

If the pressure of the steam on admission to the cylinder $=OA$, then the pressure, instead of being maintained at a pressure NE to the point of cut-off, E, gradually falls from A to B.

The stroke of the piston from end to end of the cylinder (which is equal to the diameter of the crankpin path) determines the internal length of the cylinder from cover to cover, which must evidently be equal to the stroke of the piston, plus the thickness of the piston, plus twice the clearance

allowed between the piston and cylinder cover, when the piston is at the end of its stroke. This clearance, which is kept as small as possible, varies from $\frac{1}{8}$ in. to $\frac{1}{2}$ in., according to the size of the cylinder.

It will be noticed that the shape of the cylinder cover must be made to conform to that of the piston, otherwise a considerable volume of steam might be wasted at each stroke, in filling unnecessarily large clearance spaces.

Cylinder Liner. Steam Jacket.—Cylinders are sometimes fitted with a separate internal barrel, called a cylinder *liner*, as shown in the sectional view of the compound engines (Fig. 126), made of hard cast iron or of steel.

Between the liner and the body of the cylinder is a space called the *steam jacket*, which is filled with steam direct from the boiler. The depth of the jacket is about the same as the thickness of metal in the cylinder. Sometimes the cylinder covers are jacketed, as well as the body of the cylinder.

Cylinder Escape Valves.—To avoid the danger of the piston bursting the cylinder cover as it approaches the end of its stroke, owing to the occasional presence of water through priming or condensation, cylinder escape valves are often fitted on the cylinder covers.

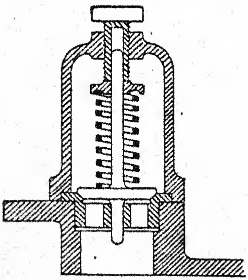


FIG. 40.

The diagram (Fig. 40) will explain the construction of these valves. The valve is of the ordinary conical kind, kept in position by a spring loaded a little above the pressure in the boiler.

Cylinder Relief Cocks (Fig. 41) are also fitted to all cylinders to drain off the water, or to blow through the cylinder with the steam, and thus clear it of water, especially on starting the engine.

Example 1.—A cylinder is 15 in. diameter, stroke of piston 25 in.; find the capacity of the cylinder, allowing an addition of 7 per cent. for clearance space. *Ans*. 4726·7 cu. in.

NOTE.—This represents the volume of steam in the cylinder at end of stroke ; the following example shows how to find the weight of this volume.

Example 2.—Find the *weight* of 4726·7 cu. in. of steam at 20 lb. pressure per square inch absolute.

By Table III 1 lb. of steam at 20 lb. pressure absolute occupies 20·08 cu. ft.

Then 20·08 cu. ft. of steam at 20 lb. pressure weigh 1 lb.

$$1 \quad \text{,,} \quad \text{,,} \quad \text{,,} \quad \text{,,} \quad \frac{1}{20{\cdot}08}\text{ lb.}$$

$$\frac{4726{\cdot}7}{1728} \quad \text{,,} \quad \text{,,} \quad \text{,,} \quad \text{,,} \quad \frac{4726{\cdot}7}{1728} \times \frac{1}{20{\cdot}08}\text{ lb.}$$

$$= 0{\cdot}1363\text{ lb.}$$

The above two examples give the volume and weight of steam used per stroke in a cylinder of the above dimensions,

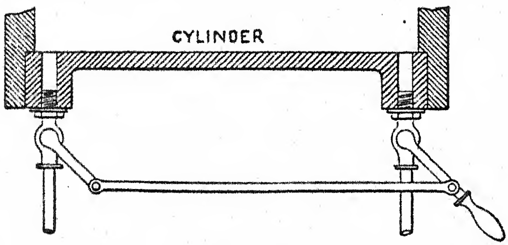


FIG. 41.

working with steam at 20 lb. terminal pressure. To find the volume or weight of steam passing through the engine as steam vapour in a given time, multiply the above results by the number of times the cylinder is filled ; in other words, multiply by the number of strokes made by the piston in the given time.

Example 3.—The engine in the above case runs at 100 r.p.m. ; find the weight of steam per hour.

In 1 stroke the weight of steam used $= 0{\cdot}1363$ lb.

,, 1 revolution ,, ,, $= (0{\cdot}1363 \times 2)$ lb.

,, 1 minute ,, ,, $= (0{\cdot}1363 \times 2 \times 100)$ lb.

,, 1 hour ,, ,, $= (0{\cdot}1363 \times 2 \times 100 \times 60)$ lb.

$= 1638$ lb.

Example 4.—Suppose it is known that the horsepower of the above engine, when working at 100 r.p.m., is 90 ; find the number of pounds of steam used per horsepower per hour. *Ans.* 18·2.

In the above example it has been assumed that the steam in the cylinder is dry. If, as is possible, the steam is wet, the *weight* of steam in the cylinder will be greater than that calculated.

Example 5.—Find the weight of 1 cu. ft. of wet steam at 20 lb. per sq. in. abs. if its dryness is 0·9.

1 lb. of dry steam occupies 20·08 cu. ft.

1 lb. of wet steam, dryness 0·9, occupies $0{\cdot}9 \times 20{\cdot}08 = 18{\cdot}07$ cu. ft.

(The volume of 0·1 lb. of water at the temperature of the steam is 0·0017 cu. ft., and may be neglected.)

The weight of 1 cu. ft. of the wet steam will thus be $\frac{1}{18{\cdot}07} = 0{\cdot}0553$ lb.

Very approximately the weight per cubic foot of steam is inversely proportional to its dryness fraction.

Pistons.—The piston is the movable plug which moves from end to end of the cylinder, under the pressure of the steam, and through which the energy of the steam is converted into the motion of the mechanism.

The piston must form a steam-tight division between the two ends of the cylinder. If it were possible to turn up a solid piston, which should so exactly fit the bore of the cylinder that it would be steam-tight, and at the same time move freely without friction, this would be a perfect piston.

In the early days of the steam engine, when steam pressures were very low, pistons were made steam-tight by coiling rope or *junk* in a groove on the rim of the piston, and this method is still adopted for pump buckets which only require to be water-tight. But for the pistons of steam cylinders a more perfect arrangement was soon found necessary. As at present made, the body of the piston is turned to an easy fit in the cylinder, and it is then made steam-tight by means of spring rings.

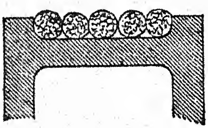


FIG. 42.

A common and simple arrangement is that of Ramsbottom's spring rings, which are simple steel or gun-metal

rings of $\frac{1}{4}$ in. to $\frac{3}{8}$ in. square section (Fig. 43). They are turned at first to a diameter a little larger than that of the cylinder they are required to fit ; and a small piece is then taken out to enable them to close up to the bore of the cylinder when in their place. They are then sprung over the piston and fitted into grooves turned in the piston rim (Fig. 43).

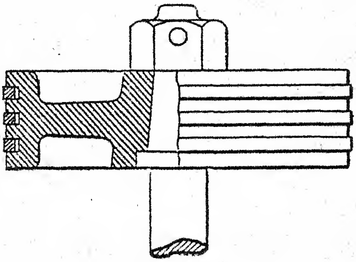


FIG. 43.

Figs. 43 and 44 are types of locomotive pistons ; Fig. 44 is fitted with two cast-iron packing rings about $\frac{1}{2}$ in. thick by $\frac{3}{4}$ in. wide, turned, cut, and sprung into position as before. The rings are sometimes placed in the same groove, and sometimes in separate grooves.

For large low-pressure cylinders pistons of the type shown in Fig. 45 are much used. The packing ring consists of one large cast-iron ring, PR, which is pressed outwards against the cylinder by means of a series of springs, S, placed behind the packing ring. For horizontal cylinders, the bottom spring is removed and a cast-iron block is substituted, which takes the weight of the piston. Instead of the small separate springs, various patent coiled springs are used in vertical engines.

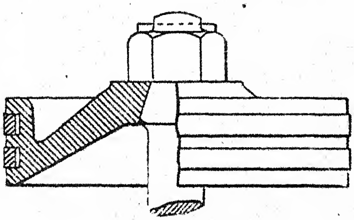


FIG. 44.

The packing ring is turned a little larger ($\frac{1}{8}$ in. per foot diameter) than the bore of the cylinder ; it is then cut through by an oblique slit and tends to spring open as wear takes place. The steam is prevented from leaking through this opening by a brass tongue piece, TP, which is fitted in another groove cut across the slit as shown. The tongue

piece is secured to a plate fastened to the back of the ring, and on one side of the slit (Fig. 46).

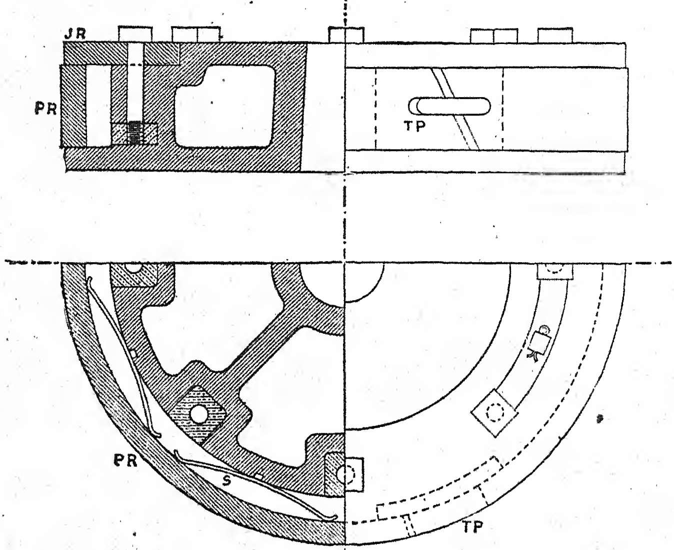


FIG. 45.
JR, junk ring; PR, packing ring; TP, tongue piece; S, spring.

The packing ring is held in its place between two flanges, one of which is cast solid with the piston, the other being formed by a loose flat ring called the junk ring, JR. The junk ring is secured to the piston by screwed bolts which screw into brass nuts inserted in a cavity left for the purpose in the body of the piston. These bolts are prevented from slacking back by a guard ring or pieces of a ring fitted between the heads, as shown.

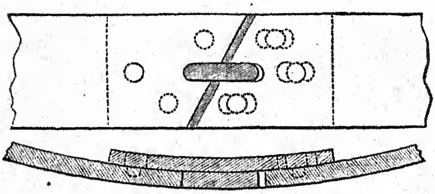


FIG. 46.

Figs. 47*a*, 47*b*, 47*c* show the Lancaster piston. The coil spring shown in Fig. 47 is in both compression and torsion when in position. The action of the coil spring is thus to keep the rings steam-tight, not only against the cylinder, but also against the junk ring and flange of the piston.

The friction between the packing ring and the cylinder should be as little as possible consistent with steam-tightness, and the piston should be as light as possible consistent with strength. Steel pistons have now become common, and by using this material the weight of the piston can be considerably reduced.

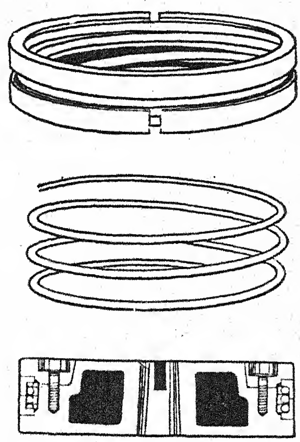


FIG. 47.

A fruitful source of loss of efficiency in steam engines is the presence of leaky pistons, the steam passing from one side of the piston to the other. Such steam is worse than wasted, as it not only does no work on the piston but acts as back pressure against it.

Piston speed.—The mean speed of the piston in feet per minute = length of stroke $\times$ number of revolutions per minute $\times$ 2.

Example.—An engine with a 3-ft. stroke makes 80 r.p.m.; find the mean speed of the piston.

$$3 \text{ ft.} \times (80 \times 2) \text{ strokes} = 480 \text{ ft. per minute.}$$

The mean speed of the piston in practice varies from about 250 ft. per minute for small stationary engines, to from 500 to 750 ft. per minute for marine engines, and in some cases it exceeds 1000 ft. per minute in the locomotive.

There has been, and is, an increasing tendency towards high piston speeds and light moving parts.

Piston displacement per minute is the space swept through by the piston at each stroke, multiplied by the number of

strokes per minute ; =the area of the piston in square feet, mutliplied by the speed of the piston in feet per minute.

Example.—Find the displacement of the piston per minute in an engine, diameter of cylinder 18 in. and length of stroke 2 ft. ; revolutions per minute, 70.

Then $(1{\cdot}5 \times 1{\cdot}5 \times 0{\cdot}7854)$ sq. ft. $\times$ 2 ft. $\times (70 \times 2)$ strokes
$= 494{\cdot}76$ cu. ft. per minute.

Piston rods are subjected to alternate pushing and pulling stresses which occur in rapid succession, and they are now invariably made of steel. The weakest part of the rod is at the screwed end which takes the nut. This part, however, is only subject to tension, and not to alternate tension and compression, for when the steam enters the cylinder underneath the piston (Fig. 38) the whole load is carried by the screwed part of the piston rod ; but on the return stroke, when the piston is descending, the stress is removed from the screwed part and comes on the tapered part of the rod and the collar.

The load to be carried by the piston rod equals the difference between the pressure on the two sides of the piston. Thus, in a condensing engine the effective pressure per square inch on the piston equals the boiler pressure by gauge, plus 15 lb. pressure at atmosphere, minus loss of pressure between boiler and cylinder, minus back pressure due to imperfect vacuum in condenser.

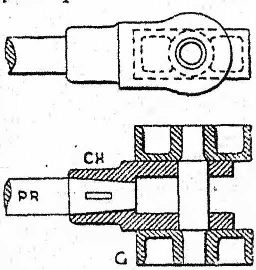


FIG. 48.

Crossheads and Guide Blocks.—The crosshead forms a head at the outer end of the piston rod, to which the connecting rod is attached by a pin passing through the crosshead. It varies very considerably in design. Guide blocks are sometimes attached to each end of the pin, on either side of the crosshead, as in Fig. 48. Another arrangement is to make a foot solid with the crosshead, which acts as a guide block, and works between guides, as shown in Fig. 49.

The blocks and guides prevent the oblique thrust or pull of the connecting rod from bending the piston rod. This can be seen by reference to Fig. 50. When the piston P is being impelled forward, so that the rotation of the crankpin is in the direction of the arrow, the resistance at the crank-

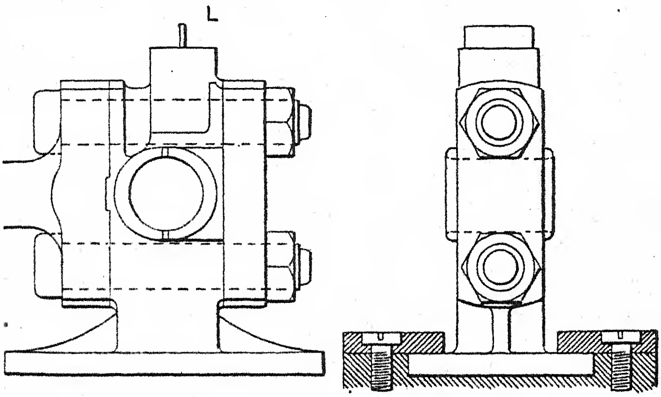


FIG. 49.

pin *c* causes a downward thrust through the connecting rod *cg*, which may be resolved into two forces, one tending to compress the piston rod and the other to bend it in the direction T, causing a downward thrust upon the guides.

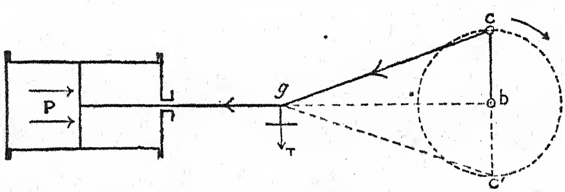


FIG. 50.

Again, when the piston is being driven back by the steam, the resistance of the crankpin at *c*′ causes a downward pull at the point *g* of the piston rod, the tendency again being to cause a downward thrust upon the guides. If the engines were reversed the whole of the conditions would

be reversed, and the thrust gT would be upwards instead of downwards. Hence the prevailing direction in which horizontal engines should run is that shown by the arrow in the figure, so that the pressure on the guides should be upon the lower rather than upon the upper guide bar ; this is especially important for the sake of efficient lubrication.

It should be noticed that when the crankpin drags the piston, as it does, for example, when steam is shut off while the engine continues to rotate, the direction of the thrust on the guides is reversed ; hence the necessity for a top and bottom guide bar under all circumstances. The amount of the thrust on the guides varies according to the angularity of the connecting rod, being greatest when the crank is at right angles to the axis of the piston rod, and being reduced to nothing at each end of the stroke ; hence the guides wear hollow in the middle, and arrangements should exist for removing the guides and truing them up.

The amount of the thrust on the guides in the middle of the stroke may be found from the following simple formula :

Maximum thrust

$$=\frac{\text{Pressure on piston} \times \text{radius of crank in inches}}{\text{Length of connecting rod in inches}}$$

Example.—Find the maximum thrust on the guides when pressure on piston at half-stroke $=20{,}000$ lb. ; radius of crank $=15$ in. ; length of connecting rod $=5$ ft.

$$\text{Maximum thrust}=\frac{20{,}000\times 15}{60}=5000 \text{ lb.}$$

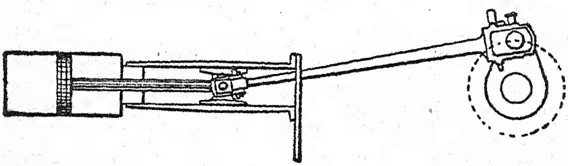


FIG. 51.

(Dotted circle = crankpin path.

When engines are required to rotate in either direction equally, as the locomotive, the surfaces in contact between the block and the guide are made equally large, as is the case in Fig. 51, with the top and bottom guide bar ; but

when the engine is intended to rotate always in one direction, or nearly so, as in the marine engine and in

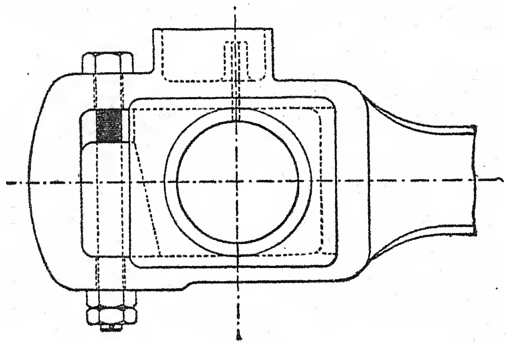


FIG. 52.

factory engines, the surface on which the thrust comes is made sufficiently large, while the opposite surface may be

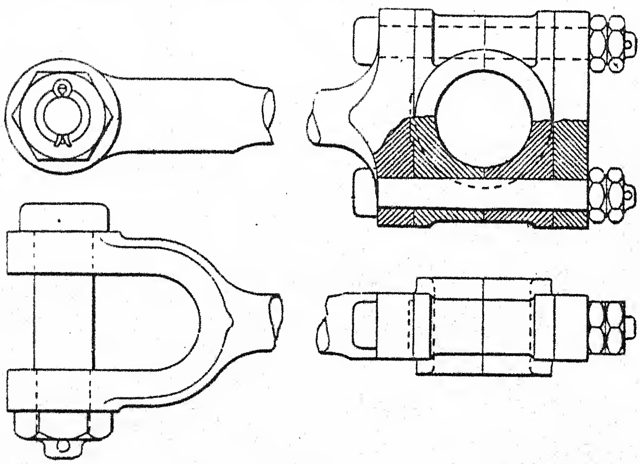


FIG. 53.

much reduced, as is the case with the *slipper, or shoe guide* (Fig. 49), the prevailing direction of the thrust being taken on the largest surface of the block.

The Connecting Rod.—The connecting rod connects the crosshead with the crankpin, and by its means the reciprocating or to-and-fro motion of the piston is transformed into the rotatory or circular motion of the crankpin.

The length of the connecting rod, which is measured from the centre of the crankpin to the centre of the cross-

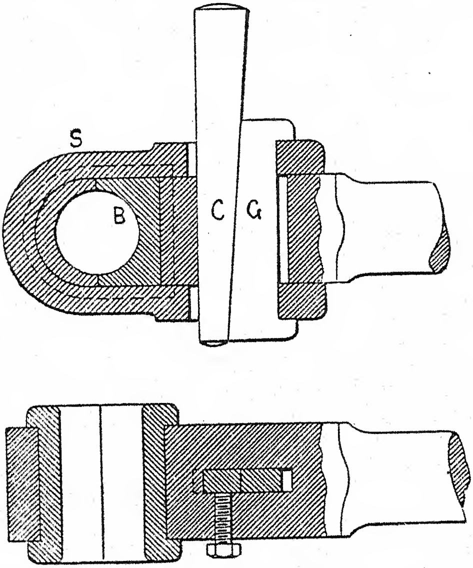


FIG. 54.

S, strap ; G, gib ; C, cotter.

head pin, varies from two to three times the length of stroke of the engine. By the *stroke* is meant the distance travelled by the piston from one end of the cylinder to the other, which is equal to the diameter of the crankpin path, or to twice the length of the crank arm.

Fig. 53 is an illustration of the marine type of connecting rod.

Fig. 54 shows a 'strap, gib, and cotter' arrangement for a connecting rod end.

Relative Positions of Piston and Crankpin.—When the piston P is at either end of the stroke, the centre line of the connecting rod BC and of the crank *o*C lie on the axis of the cylinder produced (see Figs. 55 and 56), and the crank is then said to be on its *dead centre* ; for if the

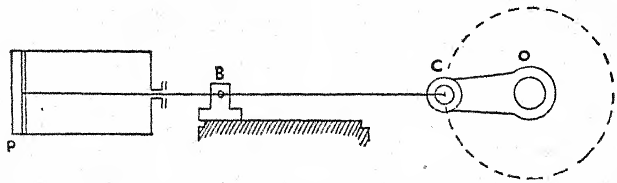


FIG. 55.

engine comes to rest in this position, it will remain at rest even when steam is admitted to the cylinder, because the pressure of the piston is felt merely as a thrust on the crank-shaft main bearing, and it has no tendency to cause the crank to rotate. In such a case it is necessary to 'bar' the engine round by the flywheel till the crank has moved

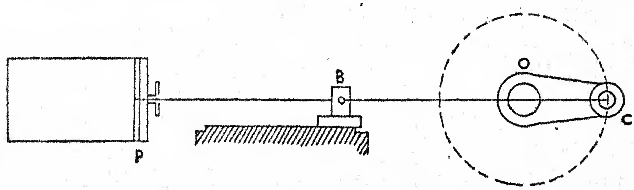


FIG. 56.

off the dead centre, before admitting steam against the piston. There are two 'dead centres' in a revolution.

Let the dotted circle, Fig. 57, represent the path of the crankpin about the centre of the crankshaft.

If the connecting rod were infinitely long, or if we neglect the obliquity of the connecting rod, then, when the crankpin is at any position between 0° and 180°, the corresponding position of the piston is found by dropping a perpendicular

upon the diameter as shown, which diameter may be taken to represent the stroke of the piston. In such a case, when the crankpin is at 90° or one-fourth of a revolution from 0°, the piston would be in the middle of its stroke; but this is not the case in practice, because of the obliquity of the connecting rod, as will now be shown.

Since the position of the crosshead corresponds exactly with the position of the piston, we may for the present purpose suppose the connecting rod to take hold of the

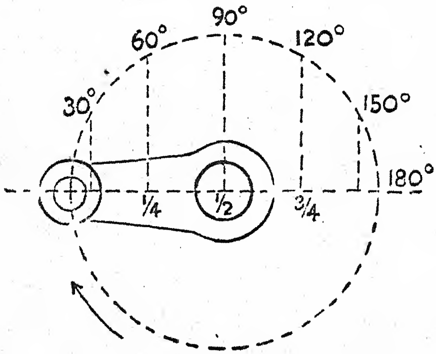


FIG. 57.

piston direct, without the intervention of the crosshead and piston rod.

First, to obtain the position of the piston for a given position of the crankpin.

In Fig. 58 let C_1C_2 be the diameter of the crankpin path, and let the length of the connecting rod be $1\frac{1}{2}$ times the stroke, namely, $1\frac{1}{2}$ times C_1C_2. From C_1, with radius equal to length of connecting rod, mark P_1, the position of the piston when crank is at C_1; also from C_2 with the same radius mark P_2, the position of piston when crank is at C_2. From any intermediate position on the circular crankpin path, and with radius equal to length of connecting rod, cut the line of stroke P_1P_2, then the intersection will give the corresponding position of the piston. Thus, when the crankpin is at C with the crank at right

angles to the line of stroke, the piston position is not at half stroke, but at some position P_0 beyond half stroke; and the shorter the connecting rod the greater the distance travelled by the piston beyond the centre of the stroke; or, the longer the connecting rod the more nearly P_0 would coincide with the middle of the stroke.

From the figure it will be clearly seen that while the crankpin rotates at a uniform velocity through the first quarter of a revolution, the piston travels at the same time from rest at P_1 a distance P_1P_0 greater than half the stroke,

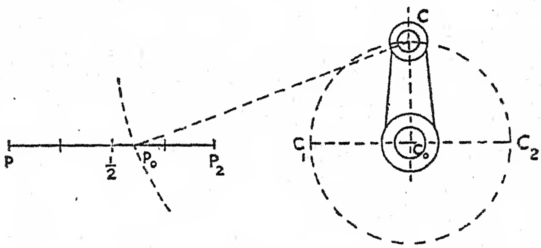


FIG. 58.

when its velocity is equal to that of the crankpin; and, during the uniform rotation of the crankpin through the second quarter, the piston travels a distance P_0P_2, or less than half the stroke, again coming to rest at P_2.

Conversely, to obtain the position of the crankpin for a given position of the piston. When the crankpin is at C_1 (Fig. 59), the piston is at P_1; and when the crankpin is at C_2 the piston is at P_2; and if the connecting rod were loose from the crankpin, and held at the centre of the crankshaft C_0 the piston would be at $P\frac{1}{2}$, namely, at the middle of the stroke. Now let the piston end of the rod remain in this middle position and move the other end of the rod from C_0, in an arc of a circle from centre $P\frac{1}{2}$ till it cuts the crankpin path at C. Then C is the position of the crankpin when the piston is in the middle of the stroke. Any other position of the crankpin for a given position of the piston may be similarly obtained.

By the term *piston speed* is meant the *mean* speed of the piston. This, however, is less than the mean speed of the crankpin ; for during one stroke of the piston the crankpin moves through a semicircular path, the length of which,

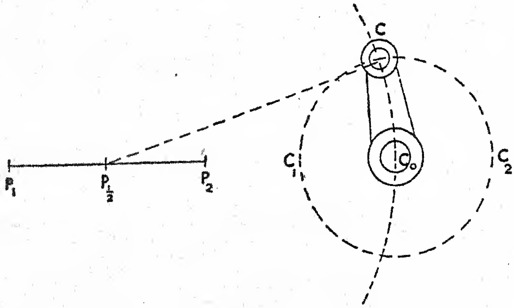


FIG. 59.

compared with its diameter or the stroke of the piston, is as $\frac{3 \cdot 1416}{2}$: 1 ; or as 1·5708 : 1.

Thus, if the mean piston speed is 1000 ft. per minute, the mean speed of the crankpin is $1000 \times 1 \cdot 5708 = 1570 \cdot 8$ ft. per minute.

By the principle of work, since the work done on the piston is the same as that done on the crankpin, and that the mean *speed* of the crankpin is 1·5708 times that of the piston, therefore the mean *pressure* on the crankpin in the direction of its motion is $\frac{1}{1 \cdot 5708}$ of the mean pressure on the piston.

Example.—In a direct acting engine the diameter of the cylinder is 17 in., and the mean pressure of the steam is 60 lb. per sq. in., the crank being 12 in. long ; what is the mean pressure on the crankpin in the direction of its motion ? (Sc. and A., 1878.)

$$\text{Then mean pressure on piston} = 17 \times 17 \times 0 \cdot 7854 \times 60$$
$$= 13{,}614$$
$$\text{and mean pressure on crankpin} = 13{,}614 \times \frac{1}{1 \cdot 5708}$$
$$= 8670 \text{ lb.}$$

Fig. 60 shows how the short connecting rod affects the position of the piston relatively to the crankpin at the two ends of the stroke. Thus crank positions A and B represent corresponding angular movements from the respective 'dead centres,' but on referring to the piston positions

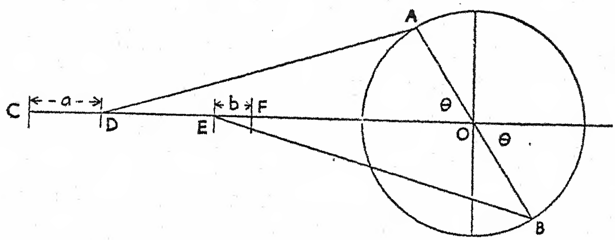


FIG. 60.

along the piston path CF we find a great discrepancy ; the distance *a* from the end C of the stroke is much greater than the distance *b* from the end F. It is necessary to allow for this in designing valve gear to cut off the steam in the cylinder at equal fractions of the stroke on each side of the piston.

Example.—Steam is cut off in an engine cylinder when the crank has travelled 120° from the dead centre both on the instroke and on the outstroke. Find the points of cut-off, assuming the connecting rod to be four cranks long.

Let OC be the length of the crank (Fig. 61). Describe the crank

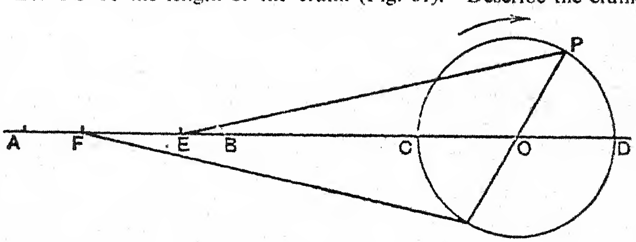


FIG. 61.

circle CPD. Make AC=4 times OC, and BD=4 times OC. Make the angle COP=120°. With centre P and radius=4 cranks, find E the position of the piston. Then the cut-off$=\frac{AE}{AB}=0{\cdot}79$ by measurement. Similarly the cut-off for the return stroke$=\frac{BF}{AB}=0{\cdot}71$.

To place the Engine on a Dead Centre.—Turn the engine round until the crank makes at A an angle of about 45° with the centre line of the engine. Make a centre dot on the slide bar and with a short trammel T make an arc of a circle on the slideblock and put a centre dot on the bar (see Fig. 62). With a long trammel K resting on a fixed point, make a mark G on the flywheel.

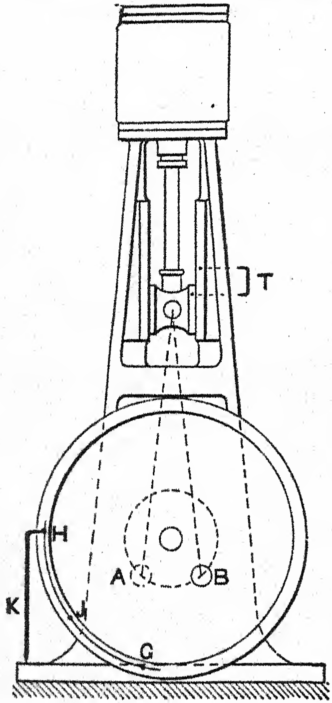


FIG. 62.

Next turn the engine back so that the crank passes through the dead centre to position B and the side block comes to the same position as before, which can be determined by the short trammel T.

With the long trammel K resting on the same fixed point, make a second mark H on the flywheel. Bisect GH at J and place J opposite the end of the long trammel. The engine will then be correctly set on one dead centre. In a similar manner the engine can be set on the other dead centre.

CHAPTER VII

THE SLIDE VALVE

Before explaining the action of the valve, it will be helpful to the student to have a clear idea of the actual shape of the cylinder face. This is shown in the following diagram, Fig. 63, where the slide jacket cover and slide valve are

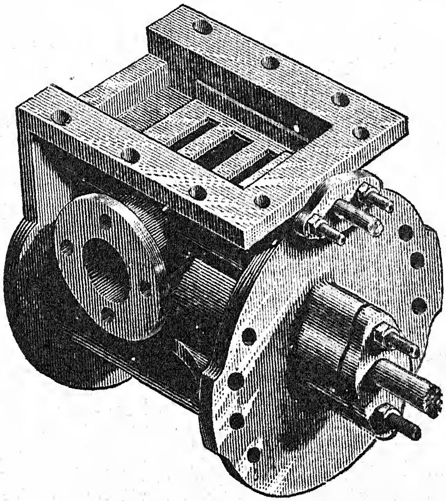


FIG. 63.

removed so as to expose to view the long rectangular-shaped ports in the cylinder face in the upper part of the diagram. Three rectangular openings are shown ; the middle port, which is the exhaust port, is wider than the other two. It is a passage leading direct from the cylinder face to the outside of the cylinder, one end of the passage being the rectangular opening, called the exhaust port, and the other end the circular opening with a flange, shown in the figure,

to which the exhaust pipe may be bolted. The two other ports are the steam ports—one leading to one end of the cylinder, and the other to the other end.

The slide valve is shaped somewhat like a hollow, rectangular, inverted dish ; the edges of the dish, constituting the face of the valve, are planed and scraped to a perfectly true plane surface, and this works on a similarly prepared surface on the cylinder face.

The following diagram will explain the action of the slide valve. We will first take the simplest form of valve in which the edges of the valve are exactly the same width as that of the steam ports.

Fig. 64 shows such a valve in its central position completely closing both steam ports. The position of the piston at the same moment is at the end of the stroke, ready to commence a new stroke. The piston is connected to the crankpin C of the crank OC moving about the centre O of the crankshaft (shown out of its correct position for the sake of convenience), and the slide valve is connected to the pin E of the smaller crank OE moving about the same centre. The centre E is really the centre of the eccentric ; but, as will be explained later on, the action is the same as though OE were a little crank. The dotted circles representing the paths of the crankpin C and of the centre of the eccentric E have their diameters equal to the stroke of the piston and valve respectively, and the positions C and E of these centres are correctly placed relatively to the positions of the piston and valve in Fig. 64. The smallest movement of the shaft about its centre O in the direction of the arrow will cause the valve by its connection with E to uncover the left-hand port and admit steam against the piston.

Suppose the shaft to have described one-fourth of a revolution from the first position, then the new positions of the pins C and E, and of the piston and valve respectively, are shown in the Fig. 65. The distribution of the steam may be also followed by referring to the arrows, the steam being admitted from the boiler on one side of the

piston, and on the other side exhausted into the air, or a condenser, by passing out through the hollow part of the valve into the exhaust passage.

As the piston continues to travel towards the end of its stroke, it will be seen, by following the movements of C and E, that the valve returns to its middle position, and again just closes the port as the piston reaches the end of

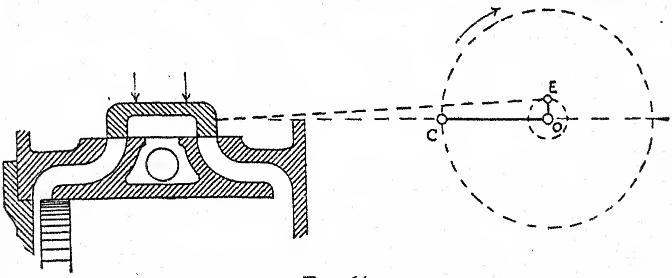


FIG. 64.

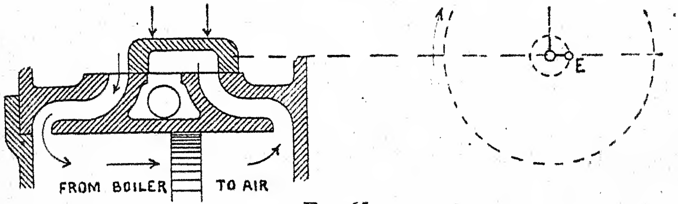


FIG. 65.

the cylinder. The valve then uncovers the right-hand port and the distribution of steam is reversed.

The valve which we have so far described has two important disadvantages :

(1) It admits steam to the cylinder throughout the whole length of the stroke of the piston. The waste of steam involved in not cutting off the supply at an early point of the stroke, and using it expansively, has been already pointed out.

(2) It opens the ports to steam and exhaust just *after* the piston moves forward on its return stroke instead of just *before* it commences to return.

These disadvantages are overcome in two ways: (1) by adding *lap* to the valve—that is, by extending the width of its face—and (2) by giving it *lead*—that is, by causing it to move forward so as to open the port just before the piston reaches the end of its stroke.

Definitions of Lap and Lead.—The amount by which the valve overlaps the edges of the steam port *when at the middle of its stroke* is called the *lap* of the valve.

The amount by which it overlaps the outside edges is called the *outside lap*.

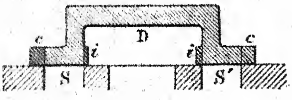


FIG. 66.

The amount by which it overlaps the inside edges is called the *inside lap*.

Thus, in Fig. 66, the lightly shaded part shows the valve with no lap. The darker parts show the addition of outside lap *cc* and inside lap *ii*, by increasing the width of the face from the width of the port S to the width *ci*.

The amount of opening of the port for the admission of steam *when the piston is at the beginning of its stroke* is called the *lead* of the valve (pronounced *leed*). Thus the opening *b* (Fig. 67) is the lead of the valve, if the piston at this moment is at the beginning of its stroke.

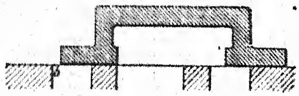


FIG. 67.

It will be noticed that, the inside lap *i* being less than the outside lap *c*, the lead to the exhaust port is greater than that to the steam port, which permits of a ready escape to exhaust.

When a valve has no lap, it moves on each side of its middle position, in order to open the steam port fully, a distance equal to the width of the port.* In other words, the radius OE (Fig. 64)=width of port. But, when lap is added to the valve (Fig. 68), the distance moved on each side of its central position must be increased, if the port is to be fully opened, to the width of the port plus the lap.

* In practice the maximum opening to steam is only about two-thirds of the port width, the full opening being taken for exhaust.

Hence the radius OE (Fig. 68) representing the eccentricity of the eccentric or the half travel of the valve=width of port+lap.

Let the piston be situated at the beginning of the stroke (Fig. 69) ; then, to admit steam to the cylinder, the valve must be moved forward from its middle position *a*, past the edge of the port *b*, until it has opened the port by a distance equal to the lead required, namely, *bc*. To accom-

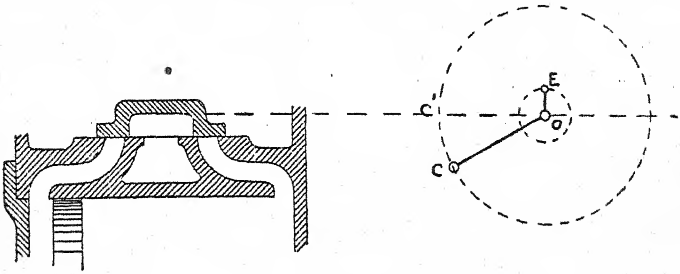


FIG. 68.

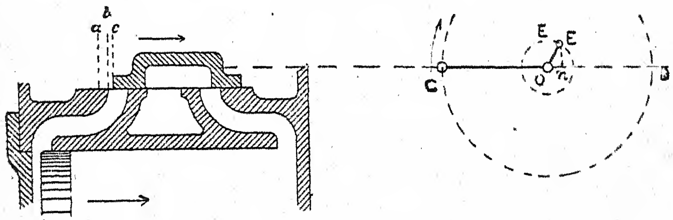


FIG. 69.

plish this, the centre of the eccentric E must be moved forward to some position E′, making an angle COE′ with the crank greater than a right angle. To find this position : From the centre O on the centre line CD set off O*n* equal to *ac*—that is, equal to the lap plus the lead—and from *n* raise a perpendicular *n*E′ to cut the circular path of the eccentric centre. Then E is the position required, and OE′ produced is the centre line of the eccentric (see also Fig. 68).

But the piston is assumed at the end of its stroke, there-

fore OC is the position of the crank, and we now have the relative positions of the crank and eccentric centre lines.

The angle EOE′ which the centre line OE′ of the eccentric is moved through *beyond* 90° *ahead of the crank* is called the *angular advance* of the eccentric (see also Figs. 70 and 74).

Example.—The width of a steam port is 1 in., the lap of the valve $\frac{1}{2}$ in., and lead $\frac{1}{8}$ in. Find the eccentricity of the eccentric, and the angular advance of the eccentric.

$$\begin{aligned}\text{Eccentricity of eccentric} &= \text{half travel of valve}\\ \text{Half travel of valve} &= \text{lap} + \text{port opening}\\ &= \tfrac{1}{2} + 1 \text{ in.}\\ &= 1\tfrac{1}{2} \text{ in.}\end{aligned}$$

Therefore, from centre o, with radius OE$=1\frac{1}{2}$ in., draw a circle representing the path of the centre of the eccentric. (Fig. 70 half-size.) Let OC be the position of the crank, and draw oE at right angles to Co. On Co produced make $oa=\frac{1}{2}$ in. and $ab=\frac{1}{8}$ in., and from b draw bE′ perpendicular to Co to cut the path of the eccentric centre in E′. Join OE′. EOE′ is the angular advance of the eccentric.

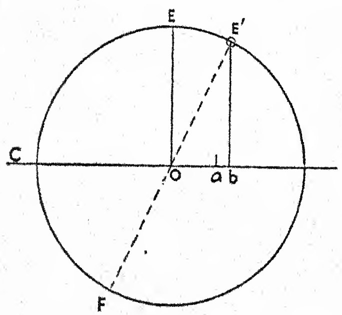


FIG. 70.

The action of the valve on the face of the ports may be easily followed by drawing the ports, and marking off the valve on the edge of a piece of paper, and moving the valve on the ports as required.

The effect of the addition of *outside lap* is :

(1) To cut off the steam at some earlier point of the stroke ;

(2) To require the eccentric to be moved forward on the shaft, which results also in an earlier opening of the exhaust port.

The effect of the addition of *inside lap* is :

(1) To close the exhaust port at an earlier point in the

stroke, producing compression of the steam at the back of the piston ;

(2) To delay the opening to exhaust.

To Set a Slide Valve.—Put the crank alternately on its two dead centres. Measure the opening of the port to steam allowed by the valve at each end of the stroke. When these are equal to the lead allowed in each case, the valve is correctly set.

The travel of a slide valve from end to end of its stroke is equal to twice the distance moved by it on each side of its middle position $=2\times$(outside lap+maximum opening of steam port).

Example.—Find the travel of a valve having $\frac{1}{2}$ in. outside lap, maximum opening of steam port $1\frac{1}{8}$ in.

$$2\times(\tfrac{1}{2}+1\tfrac{1}{8})=3\tfrac{1}{4}\text{ in.}$$

In some cases, particularly when piston valves are used, the steam is admitted by the inside edges of the valve and exhaust takes place at the outside edges. In this case the angle of advance of the eccentric is found as before, but the position of the eccentric is now exactly opposite to that for outside admission. Thus the position of the eccentric (Fig. 70) would be OF instead of OE′, the crank would be ahead of the eccentric, and the angle between crank and eccentric would be COF instead of COE′.

Piston Valves.—In the case of a simple flat slide valve the back of the valve is exposed to the pressure of the steam in the steam chest, while the greater portion of the other side of the valve is exposed to the exhaust pressure. If the valve is large the resultant force pressing the valve on its face will be correspondingly large, and since adequate lubrication is almost impossible there will be considerable friction. For instance, a valve 8 in.×6 in. with 150 lb. per sq. in. on one side and 15 lb. per sq. in. on the other side will have a force of about 3 tons pressing the valve on to the port face. Taking a coefficient of friction of 0·05 with a moderate amount of lubrication, the force required to move the valve will be about 330 lb. If the stroke of

the valve is 3 in. and the speed is 240 r.p.m., the horse-power required to drive the valve will be $\frac{330\times3\times480}{12\times33{,}000}=1{\cdot}2.$

With larger valves and higher pressures, such as are used in locomotives, the horsepower required to drive the valve

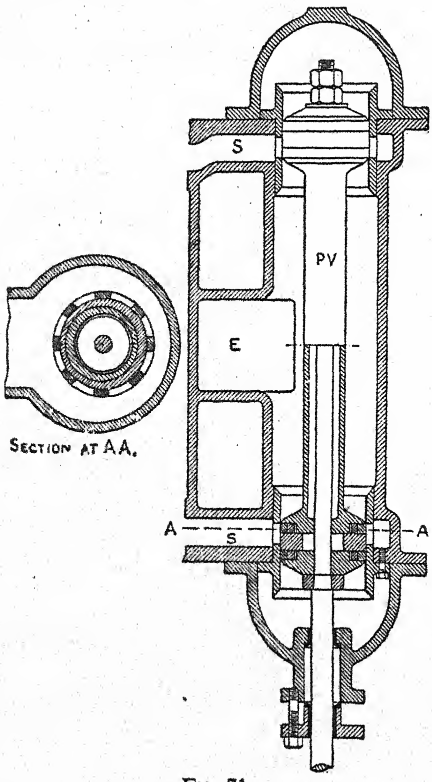


FIG. 71.

would be correspondingly higher. A relief ring at the back of the valve will shield about half the surface from pressure, but still leaves considerable forces to be transmitted to the valve rod and eccentric.

Fig. 71 illustrates the type of slide valve known as the

piston valve, so called because it consists of two pistons, each working in a short barrel, in which an opening extending right round the barrel acts as the steam port. The chief advantage of the piston valve is that it is in equilibrium, there being no pressure of the valve against the cylinder face, as with the common flat or locomotive type of valve. The face of each piston is the same as the length of the face of the flat valve, the inside and outside laps being also the same. The steam is admitted at the two ends of the valve, and exhausts into the space between the two pistons. This type of valve is suitable for high-temperature superheated steam, as being symmetrical it does not distort so readily as a flat valve. A disadvantage is that since there must be a space surrounding the valve the clearance volume is larger. The ports leading to the cylinder can, however, be shorter and with good design the increase of clearance volume need not be excessive.

Eccentrics.—Eccentrics are used when a very small to-and-fro motion is required to be derived from a revolving shaft. They are applied mostly to drive steam-engine slide valves, or pumps having a short stroke. The simplest form of eccentric is a circular solid disc called a sheave, secured to and revolving with the shaft, the centre of the disc being 'out of centre' or 'eccentric' with the centre of the shaft. This arrangement is equivalent to a small crank (Fig. 72), the length of whose arm r is the same as the distance ce between the centre of sheave and centre of shaft. This length ce is called the *eccentricity* of the eccentric. The travel of the valve is equal to twice the

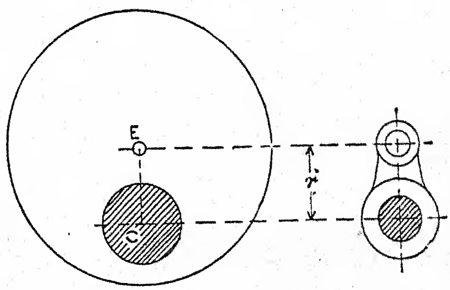


FIG. 72.

eccentricity of the eccentric. The sheave is surrounded by a thin metal hoop, or band S (Fig. 73), called the *strap*, to which the eccentric rod is attached. The rotation of the sheave H about the centre of the shaft is transmitted through the strap and rod, and results in the to-and-fro motion of the valve. The sheave may be considered as a very large crankpin, and the eccentric rod and strap as an ordinary connecting rod. The sheave rotates within the strap just as the crankpin rotates within the head of the connecting rod.

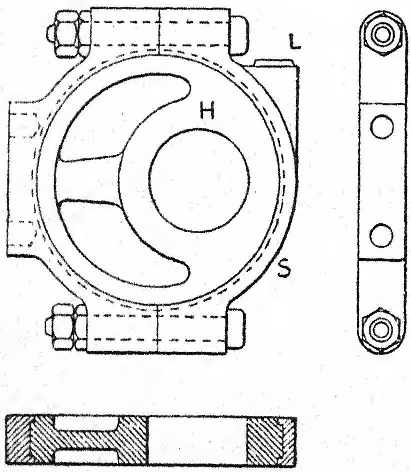


FIG. 73.

In order to get the eccentric in its place on the shaft it is usually necessary to make the sheave in halves. The halves are secured together by two bolts, not shown, which are passed through holes drilled in the sheave and secured by split cotters. The strap is also made in halves, each half having lugs to take the bolts which secure them together. A small oil cup L is cast solid with the strap.

The sheave is secured to the shaft by a key fitting in a keyway cut in the shaft and in the sheave.

The method of fixing the eccentric on the crankshaft so that it may have the correct angular advance relatively to the crank is shown in Fig. 74, in which the letters of reference O*b* and E′ correspond with the same letters in Fig. 69. OE′ is the centre line of the eccentric sheave, which has been found as explained on p. 89, and, the keyway having been marked on the shaft in the correct position, it is cut

out to receive the rectangular key which secures the sheave to the shaft.

Reversing Gear—The Link Motion.—Not the least impor-

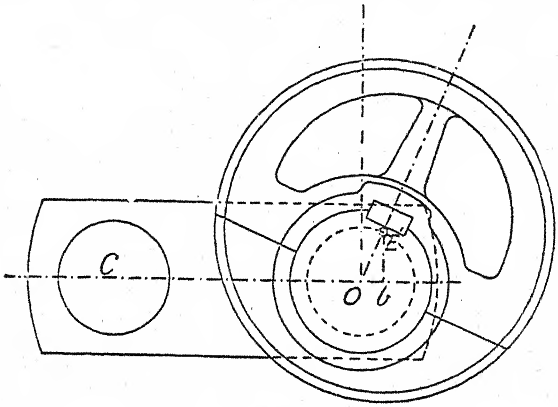


Fig. 74.

tant quality possessed by the steam engine is the ease with which it lends itself to the most perfect control. For by the movement of a handle the massive engines of a steam-

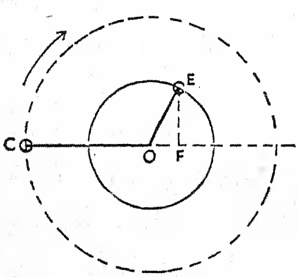


Fig. 75.

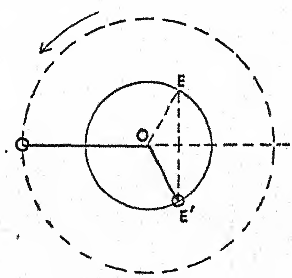


Fig. 76.

ship running at a high speed may be stopped quickly and as quickly reversed. The following simple diagram will explain the principle of reversing gears.

It has been shown that when the crank is in some position OC (Fig. 75), the centre line of the eccentric will be in a

direction OE ahead of the crank, the direction of rotation being shown by the arrow.

But suppose we wished to reverse the engine—in other words, to change the direction of rotation, as in Fig. 76—then, unless we have some means of shifting the eccentric from E to E′, the engine will not reverse, but will only rotate one way.

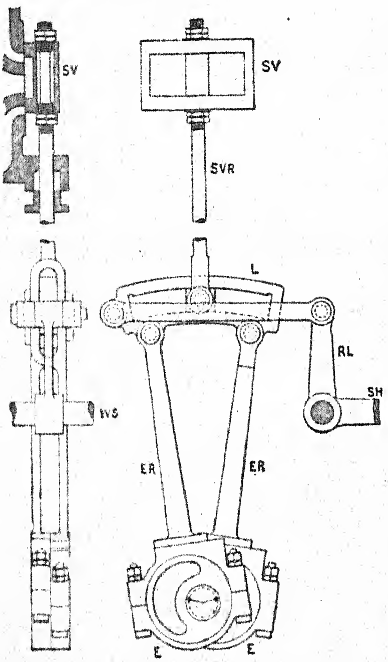


FIG. 77.

SV, slide valve; SVR, slide-valve rod; L, link; RL, reversing lever; SH, starting handle; WS, weigh shaft; E, eccentric; ER, eccentric rod.

This difficulty is easily overcome by the *link motion*, which is one of the most common methods of reversing, and it is done in the following way: Two eccentrics are used, one having its centre at E, and the other at E′ (Fig. 76), and by means of the link (Fig. 77) we have the power to use which eccentric we please, and to throw the other out of gear; hence the engine can be made to rotate in either direction with the greatest ease. Each eccentric is attached by a rod to one end of the slotted bar or link shown in Fig. 77, and the link is moved transversely by the levers so as to bring the slide valve under the influence of either eccentric as required. The slide valve is attached by a rod to a little block which fits in the slot of the link, so that any movement of the link in the direction of the axis of the valve rod affects the position of the valve.

When the block is in the middle of the link, the valve is influenced equally by both eccentrics, with the result that the engine will not run in either direction. The nearer the block is to its mid-position in the slot, the less is the travel of the valve and the earlier the steam is cut off in the cylinder. The link motion is therefore useful, not only for reversing, but as an arrangement for working the steam expansively in the cylinder by varying the point of cut-off.

For locomotive reversing gears, the Stephenson link motion has been largely replaced by Walschaert's reversing gear, in which part of the motion of the valve is obtained from the engine crosshead and the remaining part of the motion from an eccentric at right angles to the crank. A description of this gear is given in *Ripper's Steam Engine Theory and Practice*, p. 276.

CHAPTER VIII

DROP VALVE GEAR

Much of the loss by initial condensation in steam engines operated by slide valves is due to the fact that the steam enters by ports which have previously been cooled by the exhaust steam leaving by the same ports and by re-evaporation of a film of condensed steam from the port surfaces during exhaust. If separate ports are used for steam and exhaust respectively at each end of the cylinder, opened and closed by separate valves, the following advantages are obtained :

(*a*) Improved economy due to reduced condensation losses.

(*b*) Reduced clearance volume, since the ports are very short.

(*c*) Cut-off can be adjusted separately, without affecting release or compression.

In the earlier type of engine using separate steam and exhaust valves, Corliss valves were used. A section of the cylinder of a Corliss engine is shown in Fig. 78.

A number of these engines are still working with saturated or only slightly superheated steam, but with the increasing use of higher superheats it was found that sticking of the Corliss valves and consequent irregular working resulted.

Engines of moderate speed which are now in use with superheated steam are fitted with drop valves.

Fig. 79 shows a simple type of drop valve which is lifted quickly off its seat by the actuating gear at the point of admission. At the point of cut-off the gear is ' tripped ' and the spring returns the valve to its seat quickly. Steam enters at A and is passed to the cylinder at B. This simple type of valve is impracticable, as the valve would have to

be lifted against the full pressure of the steam and the operating gear would be subjected to very heavy loads.

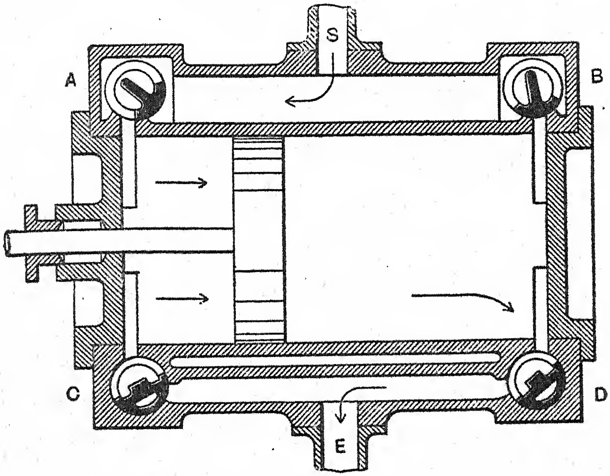


FIG. 78.

For instance, if the diameter of the valve is 6 in. and the net steam pressure is 150 lb. per sq. in., the load on the valve would be nearly 2 tons.

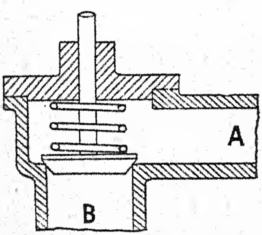


FIG. 79.

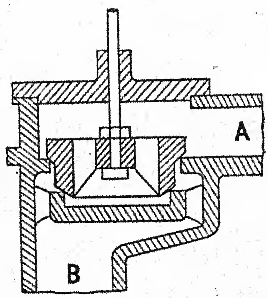


FIG. 80.

The principle of the balanced double-seated type of valve, similar to that used in practice, is shown in Fig. 80.

It will be seen that the unbalanced force on the valve is very small and, since the valve is double-seated, only half the lift is required for the same area of opening as a single seated valve.

Fig. 81 shows sections of an engine with drop valves and actuating gear as fitted by Marshall, Sons & Co. to their horizontal types of mill engines.

The steam valve 3 is lifted off its seat by the pivoted lever 18 acting on a roller 5 on the valve spindle. An eccentric 9 on the side shaft 10, which is driven at the same speed as the crankshaft, oscillates the rod 11 and the link 12. One end of the bell-crank lever 14, pivoted on 12 and moving with it, presses downward on the end of 18 and lifts the valve. When the other end of 14 reaches the fixed surface 16 the end in contact with 18 moves to the right and out of contact. The spring 7 then returns the valve quickly to its seat and cut-off takes place. The surface 16 can be raised or lowered by an eccentric 17, operated by the governor. If 16 is raised, cut-off takes place earlier, and vice versa. In order to avoid the shock of the valve striking its seating, a dashpot 6 is fitted which compresses air underneath its piston, the rate of release of the air being regulated by the small valve 8. A spring (not shown) returns the lever 14 to the position for again making contact with 18 on the upstroke of 11.

Since the points of release and compression are kept constant, whatever the cut-off may be, a simpler arrangement for actuating the exhaust valve can be used. In this case a cam 28 operates a bell-crank lever 26, and this, through the rod 25, actuates a second bell-crank lever 24, one end of which lifts the exhaust valve 19. By a suitable arrangement of the levers the valve is arranged to lift quickly, remain open for the period of exhaust, and return quickly to its seating without shock.

One difficulty with this type of valve is that unequal expansion may occur at high temperatures, so that one part of the valve seats itself while the other part is slightly off its seat, thus causing leakage of steam. In order to

overcome this either the valve or its seating is made slightly flexible in some cases.

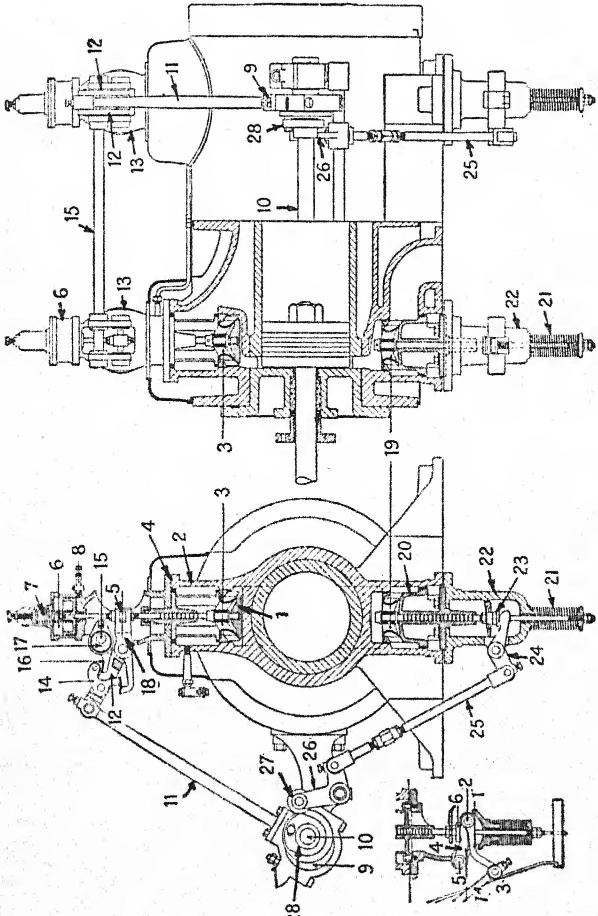


FIG. 81.

The Uniflow Engine.—A further development of the idea of using separate steam and exhaust valves is to allow the

piston to act as an exhaust valve. Fig. 82 shows how this idea is applied in practice.

Steam is admitted by a drop valve to one end of the cylinder and is cut off early in the stroke. Towards the end of the stroke the piston uncovers a ring of ports in the centre of the length of the cylinder and exhaust occurs. A great advantage of this is that the ends of the cylinder

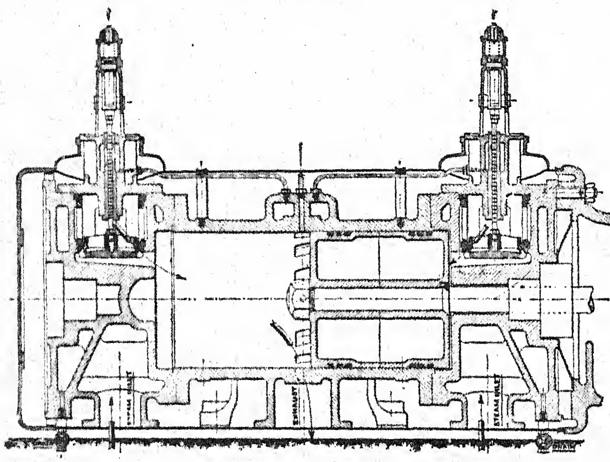


FIG. 82.

are not cooled by exhaust steam flowing back to the exhaust valve and actual tests show that the engine is very economical in steam consumption. It has, however, certain drawbacks from the point of view both of construction and of operation.

(1) Since the piston must not uncover the exhaust port until near the end of the stroke, and the length of the port is about 0·1 of the stroke, the length of the piston must be 0·9 of the stroke and so the internal length of the cylinder must be nearly twice the stroke. This increases the cost of the engine.

(2) On the return stroke the piston closes the exhaust

port at 0·1 of the stroke, and since the clearance volume is small, the final compression pressure will be very high unless the back pressure is very low.

Example 1.—Clearance volume=0·04 of stroke volume.
Compression begins at 0·1 of the return stroke.
Back pressure=3 lb. per sq. in. abs.
Compression according to law pv=constant.
Initial volume=0·94. Initial pressure=3.
Final volume=0·04. Final pressure=p.

$$p \times 0{\cdot}04 = 3 \times 0{\cdot}94$$

$$p = 70{\cdot}5 \text{ lb. per sq. in. abs.}$$

Example 2.—Data as before, but back pressure=16 lb. per sq. in. abs.

$$p = \frac{3 \times 0{\cdot}94}{0{\cdot}04} = 376 \text{ lb. per sq. in. abs.}$$

The actual compression curve rises more steeply than this, and the actual pressures reached would be of the order of 100 lb. per sq. in. and 450 lb. per sq. in. respectively. It will be obvious that the engine must be worked condensing (i.e. with a low back pressure) and that if the vacuum fails while the engine is running the compression pressure will rise far above the initial pressure and probably wreck the engine. Various relief devices are used, but probably the most efficient one is the use of a cam with variable lift which is operated by the pressure in the condenser. When the pressure is sufficiently low the relief valve provided is not lifted during the compression stroke, but as the pressure increases due to loss of vacuum the relief valve is lifted for an increasing portion of the stroke, thus automatically obviating an excessive rise of pressure.

CHAPTER IX

CRANKS AND CRANKSHAFTS

Cranks are used to convert the reciprocating motion of the piston into circular motion. Fig. 83 shows two views of a simple overhanging crank.

This crank consists of an arm with a boss at each end—one to take the main shaft, and the other the crankpin. The crank is secured firmly in its place on the shaft, either by keying alone, or by 'shrinking' and keying. The shrinking is done by boring out the hole a shade smaller than the shaft, then heating the crank round the hole, and thus causing the material to expand and the hole to become larger. The crank is then slipped in its place on the shaft, and on cooling it contracts and grips the shaft tightly. Forcing on by hydraulic pressure is now frequently adopted in preference to shrinking on. The crankpin is shrunk in position or forced in by hydraulic pressure, and riveted over the end as shown.

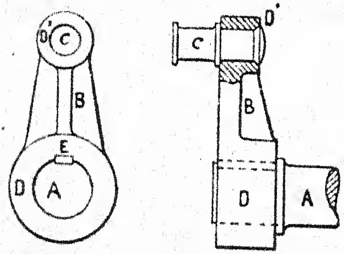


FIG. 83.
A = crankshaft; C = crankpin; B = web; D, D' = bosses; E = key.

The radius of the crank arm is measured from the centre of the shaft to the centre of the crankpin. The *throw* of the crank is equal to the diameter of the crankpin path, and to the stroke of the piston.

The following is a crank axle for a locomotive with inside cylinders, showing the cranks at right angles. The webs are here shown strengthened by wrought-iron straps shrunk

on. With such a shaft the engines will start in any position ; for, if one crank is on its ‘dead centre,’ the other is in the best possible position for starting.

Examples of crankshafts are also given in Figs. 34, 35, etc.

Tangential Pressure on the Crank pin.—The tangential pressure on the crankpin is that component of the total pressure on the pin which tends to turn it about the centre of the shaft.

To present this subject in the simplest form we will suppose the pressure of the steam on the piston uniform throughout the stroke, the connecting rod to be of infinite length, or, in other words, to act always parallel to the

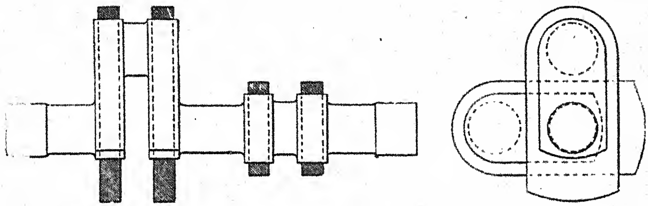


FIG. 84.

centre line of the engine, and the moving parts to be without weight.

In Fig. 85, ABC represents the path of the crankpin. Let P=the uniform pressure on the piston, and let OA the radius of the circle be chosen equal to P to any scale. When the crank is in the position OA, the pressure P acts towards the centre only, and there is no tendency to *turn* the crank about the centre, but only to press the shaft against the bearing ; hence in this position the tangential pressure is nothing. The same is true of the position OC, and OA and OC are termed the ‘dead centres.’ At the position OB of the crank at right angles to the direction of the force, the whole of the force is expended in turning the pin about O, and the tangential pressure on the pin is therefore equal to P. Between these two points A and B the tangential pressure varies from nothing at A to a

maximum at B, and again falls to nothing at C. To find the tangential pressure at an intermediate point, as E, the force P may be resolved into two forces, one acting towards the centre of the shaft, and one perpendicular to it, or tangential to the crankpin path at E. Thus at E draw the line EE′ parallel to AC and equal to the force P to scale, and by the parallelogram of forces resolve it into two forces, TE tangentially and RE radially. Then the line TE measures to scale the share of the force P acting at E, tending to turn the pin about the centre of the shaft, and RE measures also the share of force pressing the crank on the bearing. But when OE=EE′, the perpendicular EF is equal to TE for any position of E. Therefore for any point on the crankpin path, under the conditions above described, a perpendicular let fall from it upon the diameter AC represents the tangential pressure on the pin at that point.

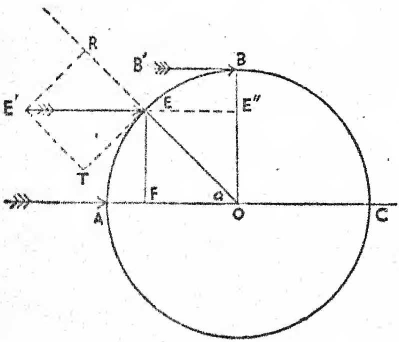


Fig. 85.

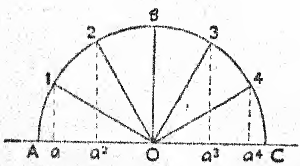


Fig. 86.

The diagram (Fig. 86) further illustrates the variable nature of the turning effort on the crankpin, the amount of the turning effort at points 1, 2, etc., being proportional to the perpendiculars $1a^1$, $2a^2$, etc., and varying from nothing at A, $1a^1$ at 1 and so on, to a maximum at B, and again gradually falling to nothing at C.

Further, it will be seen that the variation of twisting

moment on the crankshaft occurs twice in every revolution of the crank ; also that by increasing the pressure on the crankpin, either by increasing the area of the piston or the pressure of steam, the variation in the twisting moment is also increased in the same proportion.

If a pair of engines of equal power work on to one crankshaft, and the cranks are placed at angles of 0° or 180° with each other—that is, with the cranks together or exactly opposite—the twisting moments on the shaft will be double those produced by the single engine alone, also the maximum and minimum twisting moments on each crank will occur at the same time. But if the cranks be placed at right angles with each other, then, with the same engines, the maximum stress due to one engine will occur at the same time as the minimum stress due to the other engine, so that the total maximum stress will be reduced, and there will also be a much more uniform distribution of the stresses in the shaft. This will be more clearly seen by referring to the following figures.

Fig. 87 is a continuous diagram of the turning effort on the crankpin for a single engine, the value of which at

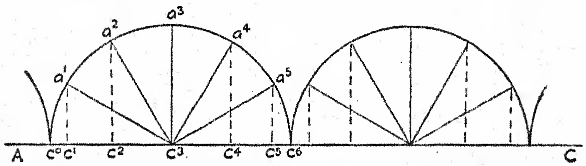


FIG. 88.

any point a^1, a^2, etc., is given by the vertical ordinate a^1c^1, a^2c^2, and so on, varying from nothing at $c°$ to a maximum a^3c^3 at a^3.

Fig. 88 shows the effect of the addition of another engine of equal power to the same shaft, when the cranks are at 0° or 180° apart. In this case the stresses are doubled throughout, varying from nothing to a maximum a^3b^3, which is twice a^3c^3 in Fig. 87.

Fig. 89 shows the effect of placing the cranks at right angles to one another, the maximum turning effort as $b^{\circ}c^{\circ}$ for one engine occurring when the turning effort of the other engine is nothing. The maximum stress is therefore never

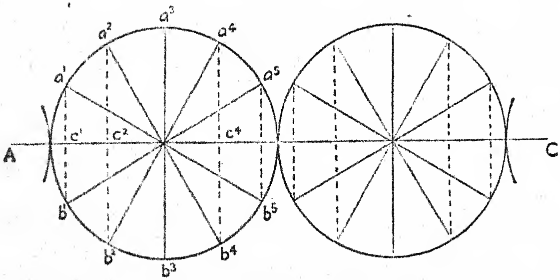


FIG. 88.

so great as twice that due to the single engine, and the minimum stress never falls below the maximum due to one engine alone. There is, therefore, a much more regular distribution of the stress.

The uniformity or otherwise of the turning effort can be

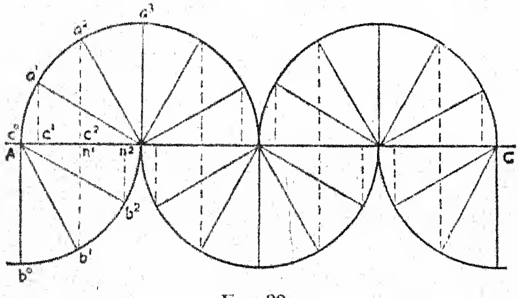


FIG. 89.

more clearly seen by setting up the ordinates from a horizontal base, as in Fig. 90. Thus, draw a horizontal line AC, and mark from A divisions Ac, etc., equal and corresponding to the divisions on the semicircumferences, Fig. 89.

Then from these divisions (Fig. 90) set up $a'c = a'c'$ (Fig. 89), and cb' (Fig. 90) $= b'n'$ (Fig. 89), and so on, and join the free ends of the lines. Then the total breadth of the figure gives the combined turning effort on the shaft. The variation in the stress may be still more conveniently represented

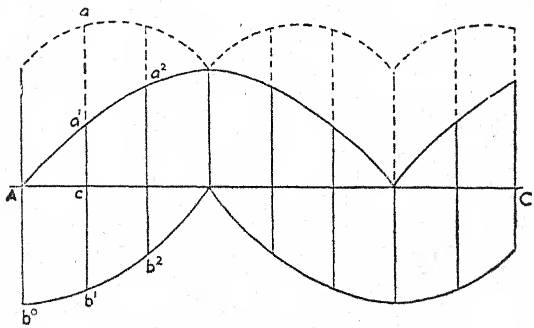


FIG. 90.

by constructing the whole figure above the line AC as shown by the dotted parts: thus cb' is set off above $a' = a'd$, and so on; and the tops of the ordinates are joined by a free curve. The nearer this curve becomes to a horizontal line, the less the variation in the twisting stresses. For the treatment of this subject, when account is taken of the varying pressures of the steam throughout the stroke, the obliquity of the connecting rod, and the weight of the moving parts, the student is referred to the Author's work on *Steam Engine Theory and Practice*.

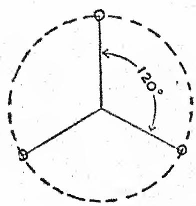


FIG 91.

Crankshafts having three cranks usually placed at 120° still further distribute the stresses, and cause a still more regular and uniform motion of the shaft.

The variable character of the tangential or turning pressure on the crankpin is due to three causes:

(1) The communication of the pressure to the crankpin

from a reciprocating piston through the connecting rod, the effect of which is that the tangential pressure varies from zero at the 'dead centres' to a maximum in the middle of the stroke.

(2) The expansive working of the steam by which the pressure falls from the beginning to the end of the stroke.

(3) The influence of the weight and velocity of the reciprocating piston, piston rod, and crosshead, which start from rest, and are accelerated till they acquire a maximum velocity at the middle of the stroke, in accomplishing which a large portion of the steam pressure is absorbed, and is

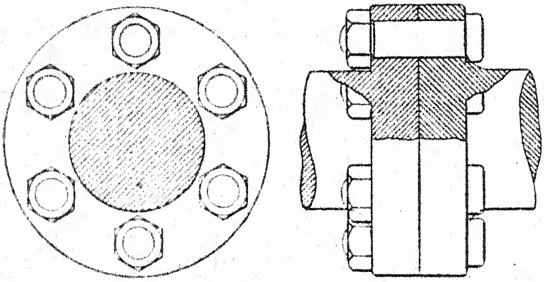


FIG. 92.

therefore not transmitted to the crankpin ; while, during the latter part of the stroke, they are again brought to rest, the effect of which is to cause a greatly increased pressure on the crankpin in addition to that due to the steam pressure on the piston.

It will be seen that the influence of the weight and of the velocity of the reciprocating parts tend to modify the variable nature of the stresses due to expansive working, and this is especially so at high speeds.

Shaft Couplings.—The above diagram, Fig. 92, illustrates the method of joining lengths of shafting together at the ends. The ends of the shafts have flanges forged on them which are turned with the shaft and butt together end to end. Holes are drilled through the flanges, and they are firmly secured by bolts as shown.

Journals.—The journal of a shaft is that part of it which fits in the bearing (Figs. 93, 94). It is of the greatest importance that the bearing surfaces of working parts should be sufficiently large. The length of the journal and of the bearing is proportioned so that the pressure per

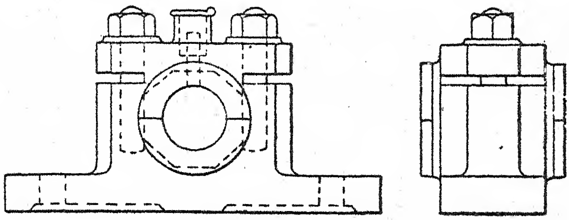


FIG. 93.

square inch on the bearing shall not be so great as to squeeze the oil out of the bearing, and so prevent proper lubrication. The length of the journal and bearing are increased for high speeds, mainly because with higher speeds more work is wasted per minute in friction and therefore more heat is generated per minute at the bearing. Now the temperature of a bearing will rise until the heat dissipated per minute (mainly by radiation) is equal to the heat generated per minute. By increasing the length of the bearing we provide more radiating surface, so that the same total amount of heat can be radiated with a smaller rise of temperature.

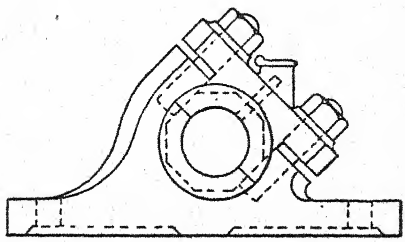


FIG. 94.

Pedestals, or Plummer Blocks (Figs. 93, 94), consist of a body which holds the brasses, and a cap which is bolted down on the brasses to keep the bearing rigid.

When the resultant of the forces acting on the shaft is

not vertical, but inclined at some angle with the vertical, the pedestal is constructed as shown in Fig. 94. The wear of a bearing tends to take place in the direction of the resultant force acting, and the joint should be at right angles to this direction. For instance, with a horizontal gas engine with the cylinder to the right the thrust due to gas pressure is horizontal and towards the left. There is also a downward force due to the weight of crankshaft, flywheel, etc., so that the resultant maximum force is inclined to the horizontal, and a bearing similar to Fig. 94 would be applicable. In the case of a horizontal steam engine, however, while the bearing would be inclined correctly for the outstroke, the inclination should be reversed for the instroke. In practice, for large horizontal steam engines separate adjustments are provided for horizontal and vertical wear.

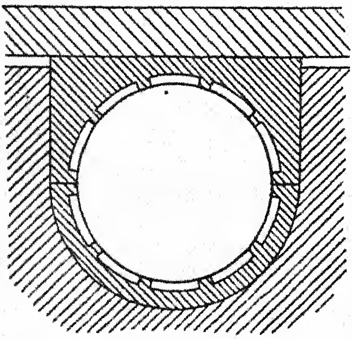


FIG. 95.

For large engines the bearings are fitted with 'white metal,' as in Fig. 95, on which shafts run more smoothly and with less friction and tendency to heat. The 'white metal' is run into grooves left for it in the brass.

CHAPTER X

CONDENSERS

The condenser is a box or chamber into which the steam is passed and condensed after doing its work in the cylinder, instead of being exhausted into the air.

The object of the condenser is two-fold, being first to remove as far as possible the effect of atmospheric pressure from the back of the piston by receiving the exhaust steam

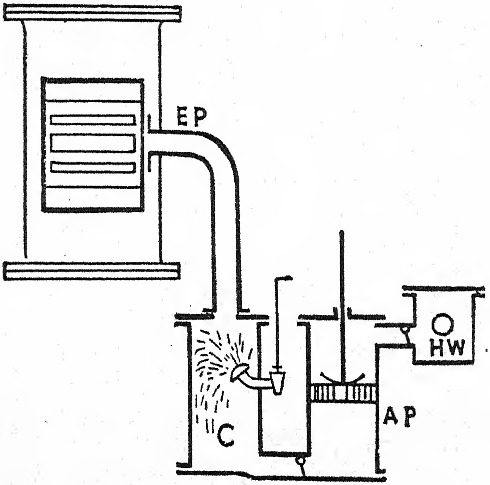


FIG. 96.

and condensing it to water, thus creating a partial vacuum ; and secondly, to enable the steam which acts on the piston to be expanded down to a lower pressure before leaving the cylinder than can profitably be done when the steam exhausts into the air.

There are two kinds of condensers, namely, the *jet* con-

denser and the *surface* condenser—the one, as its name implies, condensing the steam by means of a *jet of cold water*, and the other by bringing the steam into contact

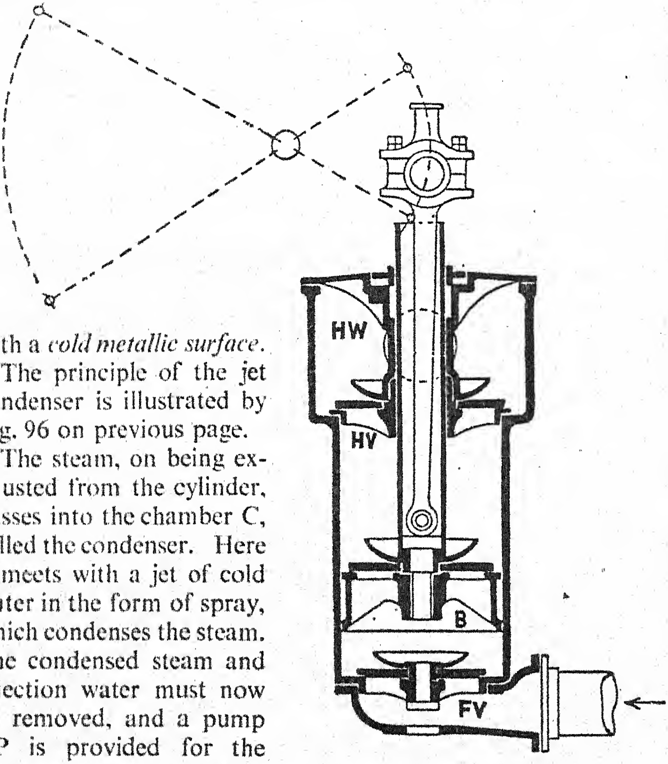


FIG. 97.

HW, hot-well ; B, air-pump bucket ; HV head valve ; FV, foot valve.

with a *cold metallic surface*.

The principle of the jet condenser is illustrated by Fig. 96 on previous page.

The steam, on being exhausted from the cylinder, passes into the chamber C, called the condenser. Here it meets with a jet of cold water in the form of spray, which condenses the steam. The condensed steam and injection water must now be removed, and a pump AP is provided for the purpose. This pump is called the *air pump* because it removes, not only the water, but also the *air* which passes into the condenser mixed with the injection water, as well as the *vapour* which arises from the water. It is the air and vapour in the condenser which are the cause of whatever *pressure* exists therein.

The condensed steam, injection water, air and vapour, are pumped into the *hot-well* HW, and thence to waste ; and from the hot-well the water is taken to feed the boiler.

The suction valve of the air pump is called the *foot valve*, and the delivery valve is called the *head valve*.

Fig. 97 is a more complete drawing of an air pump as formerly applied to vertical marine engines.

The upward travel of the pump bucket B draws air and vapour and water from the condenser through the foot valve FV, filling the space underneath the pump bucket. On the return of the pump bucket the air and water pass through the rubber valve to the top side of the pump bucket, and are delivered to the hot-well on the upward stroke. This is a single-acting reciprocating air pump.

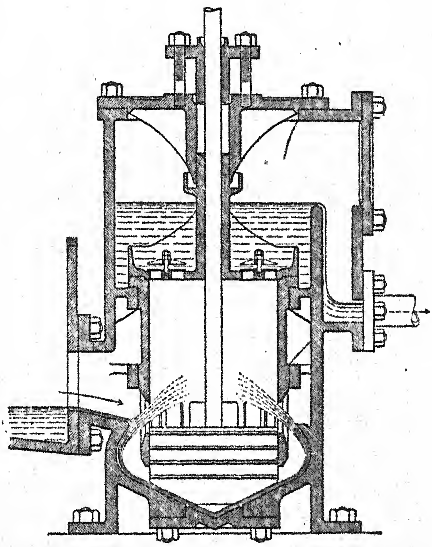


FIG. 98.

Where a vacuum of more than about 26 in. of mercury (absolute pressure of about 2 lb. per sq. in.) is required this type of air pump is unsuitable, mainly because the clearance volumes above and below the pump bucket are too large. The bucket valves and foot valves are also not easily accessible in case of breakdown. These disadvantages led to the introduction of an air pump with head valves only. This is the well-known Edwards' air pump shown in Fig. 98. Referring to the figure, when the

'bucket' is at the top of its stroke the very small clearance volume is filled with water only. Thus on the down stroke a very high vacuum is created, and when the bucket opens the ports cut in the side of the barrel, air from the condenser, however low its pressure, is forced into the upper portion. Water (condensed steam) which has drained into the lower portion of the pump during the up-stroke is driven as shown through the ports to the upper side of the bucket at high speed. Before it has had time to reverse its direction of flow and return, the bucket has ascended and trapped the water which, with the air in the barrel, is then delivered through the head valves, which remain water-sealed on the downward stroke of the valveless pump bucket. When the air pump is used to pump both the condensed steam and the air from the condenser, it is called a 'wet' air pump. In some cases it is found advisable to have separate pumps for the air and condensed steam, the air pump in this case being called a dry air pump, as it deals only with air.

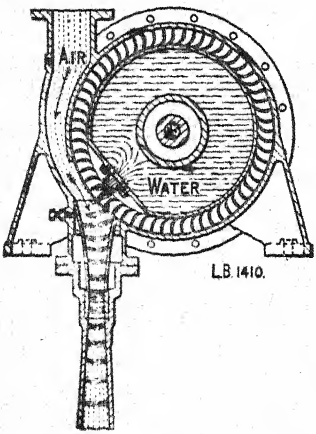


FIG. 99.

Rotary Air Pump.—Fig. 99[1] shows the principle of the Leblanc rotary air pump. Water is drawn into the central chamber by the vacuum, and passes out at the guide nozzle to the vanes of the reversed-water turbine, which serve to increase the velocity of the water. These thin discs, or plugs of water, act as pistons, and carry the air which enters at the top down the cone. The thickness of these thin discs of water is estimated to be less than $\frac{1}{100}$ in. The energy in the mixture of air and water, due to their high

[1] By favour of Messrs. Mirrlees Watson, Ltd.

velocity, is converted into pressure energy by reducing their velocity in a diverging nozzle. The mixture is discharged into the atmosphere against the atmospheric pressure. The vacuum is started by introducing priming steam for a short time, which, passing through the small annular space between the cones, draws out the air from the pump.

This type of pump is of simple construction and admits of being driven direct at a high rotative speed by a steam turbine or by an electric motor.

A still later type, which has the advantage of possessing no moving parts, is the *steam air ejector*. Referring to the diverging nozzle in Fig. 99, if instead of plugs of water a jet of steam at high velocity is introduced into the throat, air at low pressure will be entrained by the jet, and if the nozzle is sufficiently long the speed of flow is gradually reduced and the pressure correspondingly increased until the mixture of steam and air is discharged at atmospheric pressure. In practice it is found to be preferable to accomplish this in two stages, with a cooler between the two. The amount of steam required for the jets is no more than would be required to drive air pumps with moving parts, particularly when very large amounts of air have to be dealt with, and the space occupied is much less.

Both the Leblanc air pump and the steam air ejector are 'dry' air pumps, and the condensate is extracted separately.

Barometric Condenser.—A type of jet condenser which has been used for land engines is known as the barometric condenser. It depends for its action on the principle of the mercury barometer. If a tube be placed in water, as shown in Fig. 100, and all the air removed from the tube, the water will rise to a height of about 34 ft. The principle of the condenser is shown in Fig. 102. The exhaust steam enters the top of a long pipe and is condensed by a jet of water. The condensed steam and water fall down the long pipe, carrying with them the air that was present in the steam. The water must pass out of the lower end of the pipe if the latter is more than 34 ft. long, as the atmospheric pressure will not support a greater height of water

in the pipe. As the vacuum is not perfect, a height of about 30 ft. only is necessary. No air pump is required by this system as the water carries the air with it down the long pipe.

The *surface condenser* has now entirely superseded the jet condenser for marine engines as the natural consequence of the endeavour of marine engineers to increase the economy of their engines. In order to do this it was found necessary to increase the pressure of the steam used in the marine boiler, which up to 1860 was only about 30 lb. per sq. in. Up to this time the boiler feed, which was from the hot-well of the jet condenser, was practically as salt as sea water, owing to the fact that the spray of the jet condenser was a sea-water injection, the sea water and the condensed steam being in the proportion of about 30 to 1. Even with the low boiler pressures the salt in the feed water was a serious drawback, for sea water contains $\frac{1}{33}$ of its weight of solid matter dissolved in it, and, when evaporated, the solids are deposited on the boiler plates, forming a more or less thick solid incrustation. This incrustation is a bad conductor of heat, and, further, since it keeps the water from contact with the hot furnace plate, there was great danger of the plate getting red hot and the top of the furnace collapsing. To prevent the water in the boiler from becoming too much saturated with salt, it was necessary to 'blow off' a portion of the water from time to time, and to supply its place with a fresh supply of ordinary sea water. By thus blowing away to waste large

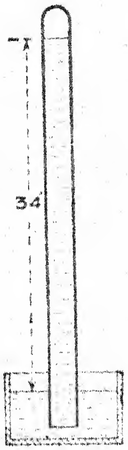


FIG. 100.

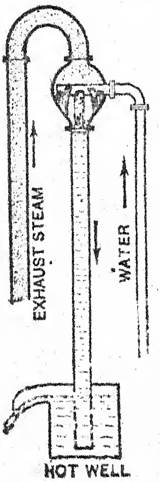


FIG. 101.

quantities of hot water, a considerable waste of heat was evidently the result.

But when the attempt was made to increase the pressure and temperature of the steam—now made possible by the introduction of steel plates for boiler construction—the difficulty arising from the presence of salt in the feed water became more serious, for with higher temperatures the solid matter is deposited much more readily, and its effects are far more mischievous. Hence the introduction of the surface condenser, which does away with the necessity of feeding the boiler with salt water, the condensed steam itself being pumped back again to the boiler as a fresh-water feed. The steam is here condensed, not by being mixed with large volumes of cold water, but by coming into contact with cold metallic surfaces.

The cold metallic surface required, by which to condense the steam, is provided by means of a large number of thin tubes, through which a current of cold water is circulated. This arrangement supplies a large cooling surface within comparatively small limits of space.

Owing to the simpler and cheaper construction of a jet condenser, this type is still used in some very large installations where an adequate amount of reasonably pure water is available, since the objection to feeding some of the water to the boiler no longer holds. If the installation is on the banks of a river the water of which is not polluted in any way a jet condenser may be used. If the quantity of cooling water available is limited a 'cooling pond' is often used, the mixture of condensed steam and cooling water being pumped from the condenser into the pond and recirculated through the condenser, the boiler also taking its feed from the pond. Usually the water in the pond does not cool quickly enough to be used again in the condenser, and 'cooling towers' are added, the water being allowed to trickle slowly over a number of wooden slats while a fan blows a current of air upwards through the tower.

The general features of one modern type of jet condenser

are shown in Fig. 102. In this type the exhaust steam enters at the side and is given a whirling motion. By means of perforated trays T as shown the cooling water descends in the form of a shower of rain, thus presenting considerable cooling surface to the entering steam. The air is withdrawn from the top at A by a 'dry' air pump and the mixture of cooling water and condensed steam is extracted at the bottom B by a pump of the centrifugal type.

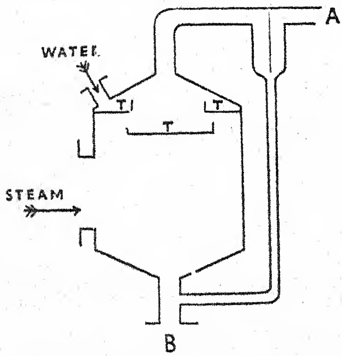


FIG. 102.

Fig. 103 shows the main features of construction of an older type of surface condenser.

The tubes are made to pass right through the condensing

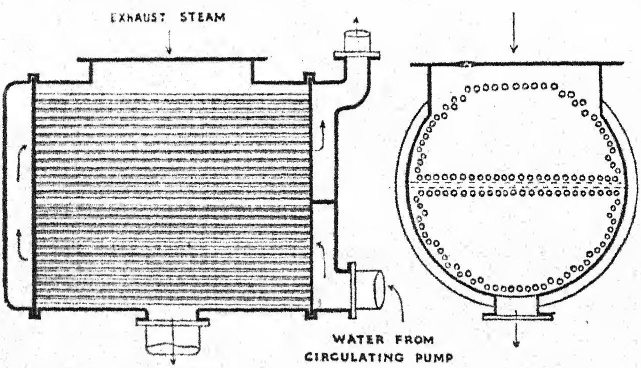


FIG. 103.

chamber, and so as to have no connection with its internal space. The steam is passed into the condenser and there comes in contact with the cold external surface of the tube, and is condensed, and removed, as before, by the air pump.

The condenser may be made of any convenient shape. It sometimes forms part of the casting supporting the cylinders of vertical engines ; it is also frequently made cylindrical with flat ends, as in Fig. 103. The ends form the tube plates to which the tubes are secured. The tubes are, of course, open at the ends, and a space is left between the tube plate and the outer covers, shown at each end of the condenser, to allow of the circulation of the water as shown by the arrows.

The cold water, which is forced through by a *circulating pump*, enters at the bottom, and is compelled to pass

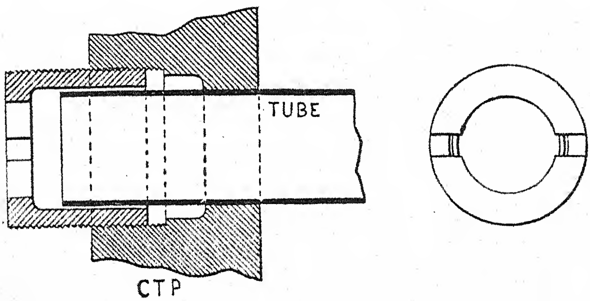


FIG. 104.
C T P = condenser tube plate.

forward through the lower set of tubes by a horizontal dividing plate ; it then returns through the upper rows of tubes and passes out at the overflow ; the tubes are thus maintained at a low temperature. The steam enters at the top of the condenser and fills the space surrounding the tubes. The tubes are made of brass, $\frac{3}{4}$ or $\frac{5}{8}$ in. outside diameter, and $\frac{1}{20}$ in. thick ; and, being thin and of good conducting material, the steam is readily condensed against the cold outer surface of the tube.

The diagram Fig. 104 shows a method of connecting the tubes to the tube plates so as to make them tight.

As the volume of condensate to be extracted at the bottom of the condenser is much smaller than the volume

of steam entering at the top the area provided at the top is much larger. (The volume of 1 lb. of steam at 2 lb. per sq. in. abs., 0·9 dry, is 156 cu. ft., and the volume of 1 lb. of water is 0·016 cu. ft.)

The rapidity with which the heat is extracted by the circulating water depends upon a number of factors, the more important of which are the following :

(*a*) The velocity of the circulating water.

(*b*) The velocity of the steam.

(*c*) The difference in temperature between the steam and the circulating water.

(*d*) The air present in the steam.

(*e*) The degree of cleanliness of the tubes.

Increasing the speed of the circulating water enables the cooling surface to condense more steam, but more power is required to drive the circulating water. Hence reduction of condenser surface may be obtained at the cost of increased pump power. The air present in the exhaust steam on entering is not condensed, and as condensation of the steam proceeds the air becomes cooler, its density becomes greater and it settles in the lower part of the condenser until the lower tubes may be so surrounded with air as to prevent them from efficiently condensing the steam, and this can only be prevented by an efficient design of condenser and an efficient air pump. The velocity of the steam as it passes over the cooling surface also affects the rate of condensation, and condensers are now generally designed to maintain the velocity of the steam by gradually reducing the size of the passage through which the gradually diminishing volume of steam passes.

The older types of condensers were usually designed to give 1 sq. ft. of surface for every 6–9 lb. of steam condensed per hour. Modern condensers condense from 15 to 25 lb. of steam per hour per square foot of tube surface. Still higher rates of condensation are possible by the use of high-water velocities.

The resistance of the tube metal to the passage of heat is very small, but a film of water 0·01 in. thick adhering to

the tube surface is equivalent to about $1\frac{1}{4}$ in. thickness of metal and a film of air 0·001 in. thick is equivalent to a thickness of metal of about 5 in.

In the type of surface condenser shown in Fig. 103 a large proportion of the steam is condensed by the upper rows of tubes. The fall of the water of condensation on the lower tubes leads to loss of efficiency (i) by employing the cooling surface to cool water instead of cooling steam, and (ii) by reducing the temperature of the hot-well feed water supply. There is also a fairly high resistance to the flow of steam through the condenser, so that the pressure at exit is appreciably lower than the pressure at entry, and consequently the vacuum available at the engine is not as good as the vacuum produced by the air pump. This is particularly objectionable in the case of steam turbines, in which a slight increase in back pressure effects an appreciable reduction in economy. Modern condensers are so designed that the steam has free access to the tubes round the shell as well as at the top and the air is extracted separately after being cooled by passing over a bank of the tubes near its exit, thus reducing the *volume* to be extracted and correspondingly reducing the size of air pump required.

It should be remembered that the condensation of the exhaust steam alone does not produce a perfect vacuum. The vacuum or absolute pressure in the condenser depends both upon the temperature in the condenser and on the air present. Assuming that no air is present, and that the temperature is 102° F., then the absolute pressure in the condenser will be 1 lb. per sq. in., because that is the pressure of the vapour which is formed at 102° F. (see Table III). If there is any air present the pressure will be still higher.

The following example shows the method of calculating the amount of cooling water required per pound of steam :

Example.—Steam enters a surface condenser at a pressure of 1 lb. per sq. in. abs., dryness 0·88, and the condensate leaves at 95° F. The cooling water enters at 65° F. and leaves at 85° F. Calculate the amount of cooling water required per pound of steam.

From Table III, total heat entering per pound of steam (from 32° F.) $=69 \cdot 5+0 \cdot 88 \times 1033=978 \cdot 5$ B.Th.U.

Heat leaving condenser per pound of condensate $=95-32=63$ B.Th.U.

$\therefore$ Heat given to cooling water per pound of steam $=978 \cdot 5-63=915 \cdot 5$ B.Th.U.

Let w = weight of cooling water per pound of steam

$$w(85-65)=915 \cdot 5$$
$$w=45 \cdot 8 \text{ lb.}$$

In order to determine the degree of vacuum in the condenser some form of vacuum gauge is required. Just as the pressure gauge represents the *difference* between the internal and external pressures, so the vacuum gauge is constructed to register the *difference* between atmospheric pressure and the pressure in the condenser. In this case, however, since the condenser pressure is less than atmospheric, the gauge reads in the opposite direction, and it is found more convenient to indicate this difference in *inches of mercury*. If one end of a tube open at both ends were dipped into a bowl of mercury and the other end were connected to the condenser by means of a pipe with air-tight joints at each end of the pipe, the mercury would rise in the tube by an amount equal to the difference between the atmospheric pressure and the pressure in the condenser. This is what the vacuum gauge indicates.

Example.—Reading of vacuum gauge $=27 \cdot 8$ in.; height of barometer $=29 \cdot 6$ in.

$\therefore$ Absolute pressure in the condenser $=29 \cdot 6-27 \cdot 8=1 \cdot 8$ in. of mercury.

Since 1 in. of mercury $=0 \cdot 49$ lb. per sq. in.

$\therefore$ Absolute pressure in condenser $=1 \cdot 8 \times 0 \cdot 49=0 \cdot 882$ lb. per sq. in.

It will be realised that, although the pressure in the condenser may not have altered, the reading of the vacuum gauge will alter as the height of the barometer alters. Thus, supposing the reading of the gauge is 29 in. when the barometer is at 30 in., if the barometer falls to 29 in. owing to atmospheric disturbances the reading of the gauge will fall to 28 in., although the pressure in the condenser is still 1 in. of mercury (0·49 lb. per sq. in.), and

such a drop in the gauge reading in this case is not due to any defect either in the condenser or the air pump. In any case, however, the gauge reading multiplied by 0·49 gives the *reduction* of back pressure due to the use of the condenser.

The gain in horsepower by using a condenser is proportional to the increase in mean effective pressure due to the reduction in back pressure. Thus if the mean effective pressure is 40 lb. per sq. in. when the back pressure is 15 lb. per sq. in. abs., a reduction of the back pressure to 2 lb. per sq. in. abs. will increase the mean pressure to $40+13=53$ lb. per sq. in., the ratio being 1·33, a gain of 33 per cent. in power.

Pumps.—The *feed pump* is used to feed the boiler, and it is required to supply a quantity of water at least equal to that evaporated and passed forward to the engine, together with leakage at safety valve, etc. ; but to provide also for emergencies it is usually made capable of supplying from 2 to $2\frac{1}{2}$ times this quantity.

The feed pump is sometimes worked from the engine direct, or from the shaft by an eccentric attached to the plunger (see Fig. 123).

With large reciprocating engines and turbines, however, the feed pump is driven separately and in many cases it consists of a multi-stage centrifugal pump.

The following diagram, Fig. 105, illustrates the construction of a simple feed pump. It consists essentially of a plunger P, of a suction valve S and a delivery valve D.

The same construction may be used for the *bilge pump*, which pumps water that accumulates in the bilge or bottom of the ship.

The action of the pump may be explained as follows : Suppose the plunger P at the bottom of its stroke, and the whole interior of the pump to be full of air. When the plunger is drawn outwards the pressure on the suction valve S will be reduced, and the air in the supply pipe will lift the valves and flow into the barrel. The pressure of the air in the supply pipe is now less than before, and

accordingly the pressure of the atmosphere on the external surface of the water forces water up the pipe to such a

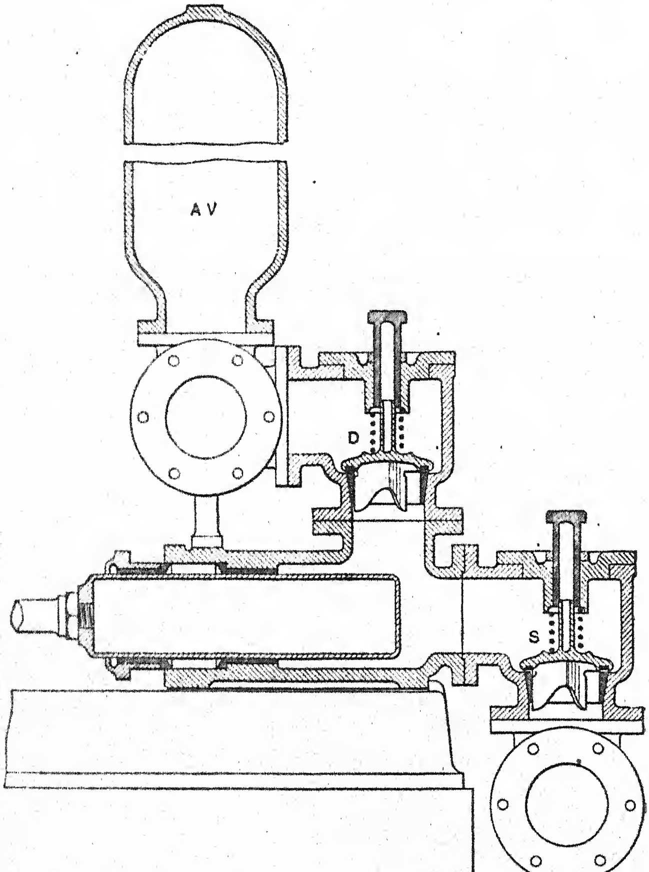


FIG. 105.—Feed pump (Messrs. Ernest Scott and Mountain).

height as to make the pressure inside the pipe balance the pressure outside. When the plunger returns the suction valve is closed by the pressure, and the air is forced out

through the delivery valve D. Each time the stroke of the plunger is repeated, the water will rise in the supply pipe until at last it reaches and fills the barrel. Now, when the plunger returns, it forces water instead of air through the delivery valve.

The height of the column of water which will balance the pressure of the atmosphere is 34 ft.; that is, a column whose weight is about 15 lb. per sq. in. In practice, however, the supply can rarely be drawn from a depth greater than about 25 ft.

The valves are prevented from rising above a certain height by the springs, shown by dots in the figure. The lift of a valve should not exceed one-fourth of its diameter, for with this lift the whole of the water which passes through the valve seating can escape freely round the edge of the valve. Any further lift is therefore useless.

Thus, when the area of opening round edge of valve and the area of the valve are equal, we have

$$\text{area round edge} = \text{area of valve};$$
$$\text{diameter} \times 3{\cdot}1416 \times \text{lift} = \text{diameter}^2 \times 0{\cdot}7854;$$
$$\text{lift} = \frac{\text{diameter}}{4}$$

Large valves are prevented from lifting so much as this, because of the excessive knocking which would result.

Air vessels AV are chambers fitted to pumps close to and beyond the delivery valve, Fig. 105. The air in the water collects in this vessel and forms a cushion or spring which enables the water to be delivered in a continuous stream instead of intermittently.

CHAPTER XI

GOVERNORS

A governor is fitted to an engine for the purpose of securing, as far as possible, a uniform rate of rotation, and preventing variation of the speed at every fluctuation in the load or the boiler pressure.

None of the governors applied to steam engines are able to accomplish this result *perfectly* ; for, being themselves driven by the engine, they cannot begin to act until a change of velocity has first occurred.

In practice, however, the governor is an invaluable adjunct to the steam engine ; for, when any change of velocity does take place, the governor acts and prevents anything more than a small alteration of speed.

The effect of the steam acting in the cylinders of a reciprocating engine or on the blading of a turbine is to produce a turning moment on the shaft. We will call this the *driving moment*. This has to overcome a *resisting moment* due to whatever is being driven. If the driving moment is equal to the resisting moment, the speed remains uniform, but if the resisting moment decreases due to a decrease of load, either the driving moment must be decreased (by decreasing the steam supply) or the speed will increase, and *vice versa*. The object of the governor is therefore to adjust the steam supply to the load with as little change of speed as possible.

The following is a description of the Watt Pendulum Governor. The study of this governor will serve to introduce the student to those principles of construction upon which this and other governors are based.

The central spindle S of the governor, Fig. 106, is made to rotate by means of a belt, or, better, by a small shaft driven from the engine shaft by bevel wheels communi-

cating with the bevel wheels at the bottom of the spindle. The spindle, arms, and balls then all rotate together, and at the normal velocity of the engine the inclination of the arms is about 30° with the vertical. If the velocity of the engine increases, due to removal of load, the balls and arms open out from the spindle, and in doing so they lift the sleeve E, which slides up and down on the spindle. This movement is communicated by levers moving about the fixed fulcrum C, to the throttle valve, by which the passage for the supply of steam to the engine is contracted, or to an expansion gear, which is also an arrangement for reducing the steam supply, and the increasing speed of the engine is thereby checked. A slot is cut in the central spindle through which a cotter or pin secured to the sliding sleeve passes. The length of this slot limits the travel of the sleeve.

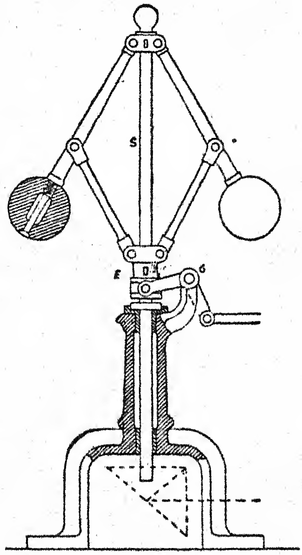


FIG. 106.

In the lower position of the balls (Fig. 107) Ac is the 'height' of the governor. If the speed increases, the balls move outward, and the 'height' decreases to ac, so that the sleeve rises by an amount proportional to Aa. If there were no resistance to the movement of the sleeve the 'height' would be given by

$$h\ (\text{in.}) = \frac{34{,}700}{N^2}$$

where N is the speed in revolutions per minute.

It will be seen from this that the simple type described is only suitable for low speeds.

Example.—If N=60, h=9·64 in.
If N=62, h=9·01 in.
Rise of A (Fig. 107)=0·63 in.
If N=240, h=0·602 in.
If N=248, h=0·565 in.
Rise of A =0·037 in.
Note that the fractional change of speed is the same in both cases.

Again, any increase in the speed of the engine causes the governor balls to move farther away from the centre, and a

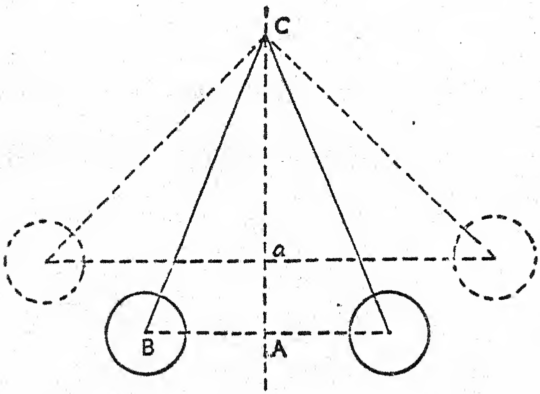


Fig. 107.

reduction in the height of the governor to take place, as from CA to Ca, Fig. 107. It is the raising of the sleeve A to a by which the governor is made to influence the throttle valve or expansion gear ; but, in order to close the throttle valve, it requires to be driven at an increased speed, and this is precisely what the governor is intended to check.

Such a governor, therefore, evidently permits of a variation in the number of its revolutions, and, therefore, also of the revolutions of the engine, between the limits due to the varying height CA of the cone of revolution. But a perfect governor would permit of no increase either in the number of its own revolutions or that of the engine ; and, although this ideal cannot be attained, still it is the aim of

designers to reduce this variation in the height of the cone as much as possible ; or, in other words, to enable the governor to lift a sufficient distance to close the valve without going through a considerable variation in speed in rising from its lowest to its highest position. The effect of the movement of the balls on the height of the cone when the point of suspension of the arms is on the centre line of the spindle is shown in Fig. 107.

When, however, the arms are suspended from points E and F (Fig. 108), not on the centre line of the spindle, and the balls rise from D to D′, the height of the cone now varies between CB and C′B′, instead of between CA and C*a* as before, the effect being to still further increase the amount of variation in height, and, therefore, in revolutions of the engine, for a given lift of the sleeve. The points of suspension E and F should, therefore, be as near the centre of rotation of the spindle as possible.

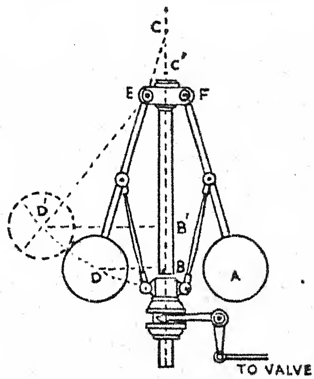


FIG. 108.

In some cases the arms are crossed, so that a very small change of speed produces a large movement of the sleeve in the endeavour to decrease the 'height.' A very small amount of frictional resistance to motion of the sleeve, however, completely upsets the working unless the balls are very heavy indeed. For this reason, and to make the governor more practicable for reasonably high speeds, in the Porter type the sleeve is loaded with an additional weight. The formula for 'height' then becomes

$$h \text{ (in.)} = \frac{34{,}700}{N^2} \cdot \frac{W+w}{w}$$

where W = central load (pounds)
w = weight of one ball (pounds)

Thus if $w=6$ lb., $W=54$ lb., then $\frac{W+w}{w}=10$ and the 'height,' and therefore the rise of sleeve for a given change of speed, is increased ten times.

The essential points of a good governor are stability, sensitiveness, and power. By *stability* in a governor is meant that there is a definite position of the governor sleeve for every given speed of the governor throughout its range of movement. Actually, the effect of friction is to give two speeds between which the governor will not act. Thus if f is the friction in pounds at the sleeve, the above formula becomes

$$h=\frac{34{,}700}{N^2}\cdot\frac{W+w\pm f}{w}$$

or

$$N=\sqrt{\frac{34{,}700}{h}\cdot\frac{W+w\pm f}{w}}$$

The plus value gives the higher speed for a given value of h and the minus value gives the lower speed for the same value of h. It will be obvious that the effect of W is to increase the *stability* of the governor.

The *sensitiveness* of a governor is measured by its degree of responsiveness to change of speed of the engine, change of speed being of course due to change of load. In other words, a sensitive governor is one the sleeve of which will move through the whole range of its position above and below its mean position for a very small change in speed of the engine.

A governor may, however, be made too sensitive, resulting in a large alteration in the steam supplied for a comparatively small change in the load. In such a case the governor tends to 'hunt,' or in other words, to close the throttle valve too much for a falling load and to open it too wide for a rising load, in which case the governor defeats its own object, and instead of regulating the speed of the engine it disturbs its regularity.

The addition of a dashpot is useful in preventing the tendency to hunt on the part of very sensitive governors.

The *power* of a governor is represented by its capacity to control the valve gear connected with it, and to overcome the resistance due to the weight and friction of such gear.

Its power is measured by the work it is capable of doing as the result of the centrifugal force of the governor in lifting itself through its full range of movement. The power of a governor depends upon the centrifugal force generated in its rotating parts, which increases directly as the weight of the balls and as the square of its velocity of rotation.

Governors of the type described above must be used with the axis vertical and on stationary engines only. Owing to the considerable inertia of the heavy central weight they are too sluggish in action where rapid fluctuations of load take place. Mainly for these reasons, this type has been obsolete for some years and is not now fitted in new engines.

Fig. 109 shows the chief features of the Hartnell type of governor, which does not depend upon gravity for its action, can be used in any position, is not affected appreciably by vibration or other movement of the engine, and possesses comparatively little inertia, since the load on the sleeve is provided by a spring, which can produce a very heavy load with little actual weight.

ABC is a bell-crank lever, pivoted at the pin B, which is fixed to the frame F, the whole revolving with the shaft R. At some arranged speed the moment of the centrifugal force on the balls A, taken about B, is just balanced by the moment of the spring pressure on the sleeve, also taken about B, when the sleeve is just about to lift. If the speed increases the moment of the centrifugal force increases, partly due to the increased speed and partly due to the increased radius of the path of each ball. As the sleeve lifts, the compression of the spring increases until the increased load on the sleeve just balances the increased centrifugal force. By suitably adjusting the stiffness and initial compression of the spring P it can be arranged that

a small increase of speed causes a large movement of the sleeve, so that the speed can be governed within fine limits. The load on the sleeve due to the spring P can be made so large that a small amount of friction does not affect the stability of the governor very much. There are variations

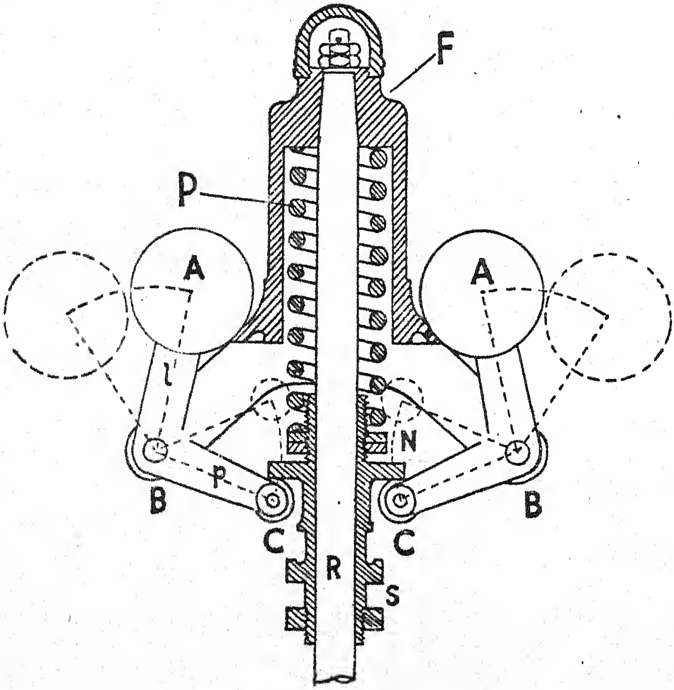


Fig. 109.

in construction and arrangement, but practically all modern governors work on this principle.

Flywheels.—The governor can limit changes of average speed of revolution between specified limits, but cannot prevent changes of speed *during each revolution*, since it cannot control the action of the steam after it has entered the cylinder. As shown in Chapter IV, the tangential

pressure on the crankpin varies during the revolution, so that at some part of the revolution the steam is doing more work than is required and at some other part it is doing less work than is required. This is the reason for fitting a flywheel to an engine.

When more work is being done than is required the flywheel stores this work in the form of energy by increasing its speed. When less work is being done than is required the flywheel supplies the extra work by giving up energy and so decreasing its speed. The flywheel thus cannot prevent change of speed during the revolution, but if its weight and diameter are sufficiently large it can store or give up a considerable amount of energy with only a small change of speed.

The amount of energy a flywheel can store is proportional to its weight and the *square* of its mean radius. It is also proportional to the *square* of its speed of revolution.

Thus $$E \propto W.r^2.N^2$$

where E=energy stored in foot-pounds
r=mean radius in feet
N=revolutions per minute.

Thus the flywheel of a high-speed engine (N large) can be smaller than that of a slow-speed engine for the same effect.

It must be borne in mind that all the moving parts of an engine and whatever it is driving have a flywheel effect. For instance, the locomotive and its wheels supply all the flywheel effect required without the actual addition of a flywheel. A single-cylinder gas engine, in which the driving moment varies enormously during the cycle, requires a much larger flywheel than a steam engine of the same power to keep the speed variation between the same limits.

CHAPTER XII

THE LOCOMOTIVE

Figs. 110 and 111 show the essential parts of a locomotive. The references to the parts are given below the figure.

It is necessary that the locomotive shall be self-contained —that is, it must consist of a boiler and an engine, and the whole machine must be placed upon one carriage. The problem for locomotive engineers is how to obtain the greatest possible power for the least possible weight. This is done by working at high steam pressures, using small boilers of great strength, and of high evaporative efficiency, and using the steam at high pressure in small cylinders in order to obtain a large amount of power with a comparatively light engine.

The engine and boiler are each bolted independently to the frame of the carriage. The frame is self-contained, and through it the whole of the stresses due to the pressure on the pistons, and the pull on the drawbar due to the load, are transmitted.

The frame is carried on wheels, one arrangement of which is shown in the figure.

The locomotive boiler is described under the heading of Boilers (Chapter XV). The engine is similar in principle to that already described on p. 64 with the addition of a reversing gear (see Chapter VII). Since the sharp blast of the exhaust from E is needed to produce the draught which causes a flow of air through the furnace, it is necessary to run the engine non-condensing, and although attempts have been made to use condensers with air coolers and a fan for the draught, none of these have proved satisfactory. Improvements in materials and lubricants have made the use of very high pressures and highly superheated

The Locomotive.

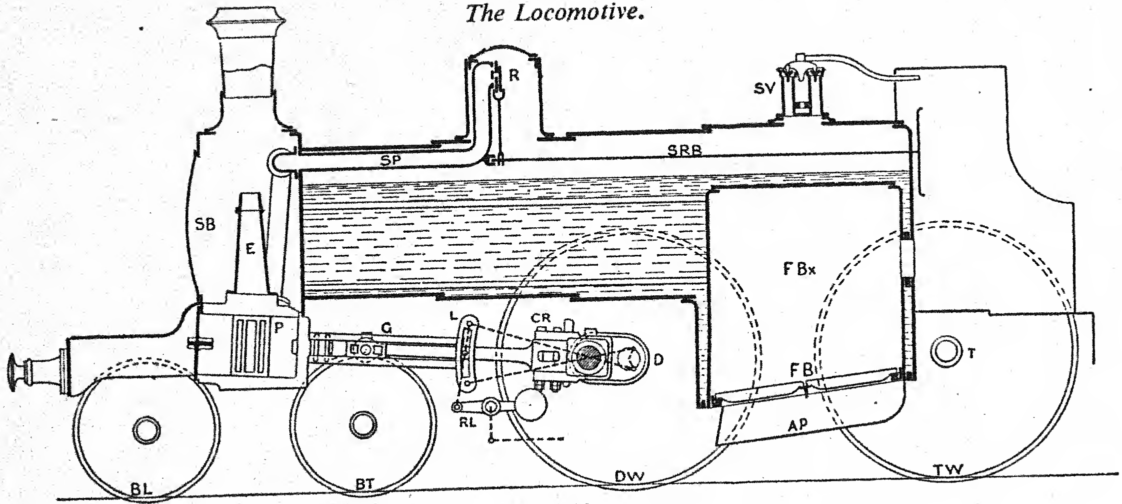


FIG. 110.

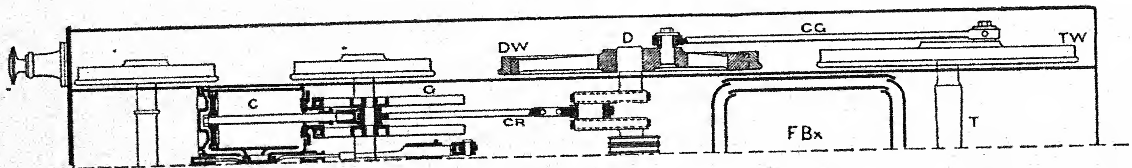


FIG. 111.

FBx, fire box ; FB, fire bars ; AP, ash pan ; SB, smoke box ; SV, safety valve ; R, steam regulator valve ; SRB, steam regulator valve bar ; SP, steam pipe ; E, exhaust pipe ; P, steam ports ; G, guides ; L, link motion ; RL, reversing lever ; CR, connecting rod ; C, cylinder ; D, driving axle ; T, trailing axle ; DW, driving wheel ; TW, trailing wheel ; BL, bogie leading wheel ; BT, bogie trailing wheel ; CG, coupling rod.

steam possible, thus reducing the steam consumption per horsepower generated very considerably. Fig. 155, p. 194, shows the type of superheater used in the majority of cases.

With comparatively low boiler pressures and saturated steam very little, if any, economy is effected by the use of compound engines, but with high pressures and superheated steam compound engines enable a larger ratio of expansion to be used with a simple type of valve gear, thus improving the economy (see Chapter XIV).

Two arrangements which are used in practice are: (1) three-cylinder compounds, using three cylinders of the same size, one of these being the high-pressure cylinder and the other two together forming the equivalent of the low-pressure cylinder. For starting, where a high tractive force is required, arrangements are made to admit boiler steam to all the cylinders, the three cranks being at 120°; (2) four-cylinder compounds, with two high-pressure cylinders and two low-pressure cylinders of large size, acting on four cranks at 90°.

Tractive Force.—The mean pull exerted by the locomotive on the train is called the tractive force. It may be calculated by considering that the work done by the tractive force in a given time is equal to the work done by the steam in the cylinders in the same time.

Let T = mean tractive force in pounds,

p = mean pressure of steam in the cylinder in pounds per square inch,

L = length of stroke in feet,

d = diameter of cylinder in inches,

D = diameter of driving wheel in feet.

Assume two cylinders of equal diameter and having the same stroke and consider one revolution of the crank axle.

Work done in two cylinders = work done on train.

$$2\frac{pLd^2\pi}{4}2 = \mathrm{T}\times\text{circumference of the driving wheel}$$

$$pLd^2\pi = \mathrm{T}\pi\mathrm{D}$$

$$\text{or} \qquad \mathrm{T} = \frac{pLd^2}{\mathrm{D}}$$

From the above equation it will be seen that the tractive force can be increased by increasing the size of the cylinders, by increasing the mean pressure, and by decreasing the size of the driving wheels. For a given size of cylinder large driving wheels give a small tractive force and high speed, and small driving wheels give a large tractive force with a slower speed. The latter condition is suitable for goods engines, and the former for passenger express engines.

Example.—The mean pressure in the two cylinders of a locomotive engine is 140 lb. per sq. in.; diameter of cylinder, 19 in.; stroke, 26 in.; driving wheel, 7 ft. 6 in. in diameter. Find the mean tractive force.

$$T=\frac{140\times 26\times 19\times 19}{12\times 7{\cdot}5}=14{,}600 \text{ lb.}$$

The actual tractive force is less than this, as internal friction has not been allowed for. Only about 80 per cent. of this is available at the drawbar connecting the engine to the tender.

The tractive force which can be applied to a train is limited by the adhesion of the driving wheels to the rails. If there is not sufficient load on the wheels, and sufficient friction between the wheel and the rail, the wheels will slip round. This occurs occasionally with greasy rails, or when starting. The use of sand on the rail by increasing the coefficient of friction increases the adhesion, and thus tends to prevent slipping.

The *maximum* tractive force must therefore be less than the adhesion of the wheel to the rail, if slipping of the wheel is to be avoided. The adhesive weight depends upon the total weight on the drivers and on the coefficient of friction between the wheel and the rail. The latter may be taken to be $\frac{3}{10}$ under good conditions. Thus, if the maximum tractive force is less than $\frac{3}{10}$ of the load on the driving wheels slipping will not take place. When the engine is starting and taking steam for the whole of the stroke, the engine will be exerting its highest mean tractive effort.

The average practice in British locomotives is to allow

the tractive effort to be from $\frac{1}{4}$ to $\frac{1}{5}$ of the weight on the drivers.

Example.—The maximum tractive force which can be exerted by a locomotive is 21,500 lb. If the coefficient of friction between wheels and rails is 0·2, what must be the minimum total load on the coupled wheels ?

$$0{\cdot}2W \doteq 21{,}500$$
$$W = 10{,}750 \text{ lb. (48 tons)}$$

Since the load which can be applied to a rail is limited by the bending strength of the rail and the resistance to crushing at the point of contact, a heavy total load must be distributed over a number of coupled axles. For instance, if in the above case the load between wheel and rail must not exceed 8 tons, then three pairs of coupled axles must be used.

The bogie wheels are attached to a frame which is pivoted to the main frame, thus allowing them to swivel slightly and guide the engine from a straight into a curved path. In some cases this purpose is served by a pair of wheels only, fitted to a swivelling frame known as a 'pony truck.'

CHAPTER XIII

THE INDICATOR

The indicator was originally invented by James Watt, and, although improved in points of detail, the main features of the instrument as devised by him are substantially retained at the present time by makers of indicators.

The uses to which the indicator is chiefly applied for steam engines are—

1. To obtain a diagram from which conclusions may be drawn as to the correctness, or otherwise, of the behaviour of the steam in the cylinder ; the promptness of the steam admission ; the loss by fall of pressure between the boiler and the cylinder ; the loss by wiredrawing ; the extent and character of the expansion ; the efficiency of the arrangements for exhaust, including the extent of the back pressure ; the amount of compression.

2. To find the mean effective pressure exerted by the steam upon the piston, from which to calculate the *indicated horsepower* of the engine.

3. To determine whether the valves are set correctly by taking diagrams from each end of the cylinder and observing and comparing the respective positions of the points of admission, cut-off, release, and compression.

When used on slow-speed gas and oil engines the indicator not only enables the indicated horsepower to be obtained but the diagram also gives useful information on the correctness of the valve setting and nature of the ignition and expansion.

Essentials of a perfect indicator are as follows :

1. The rise of the ' pencil ' must be proportional to the rise of pressure in the cylinder of the engine.

2. The motion of the drum must be an exact copy of the motion of the engine piston to a reduced scale.

The first requirement can be met by the use of a spring between the indicator piston and the cover, since the compression of a spring is proportional to the load. As, however, the height of the diagram should be not much less than an inch, a long and heavy spring would be required

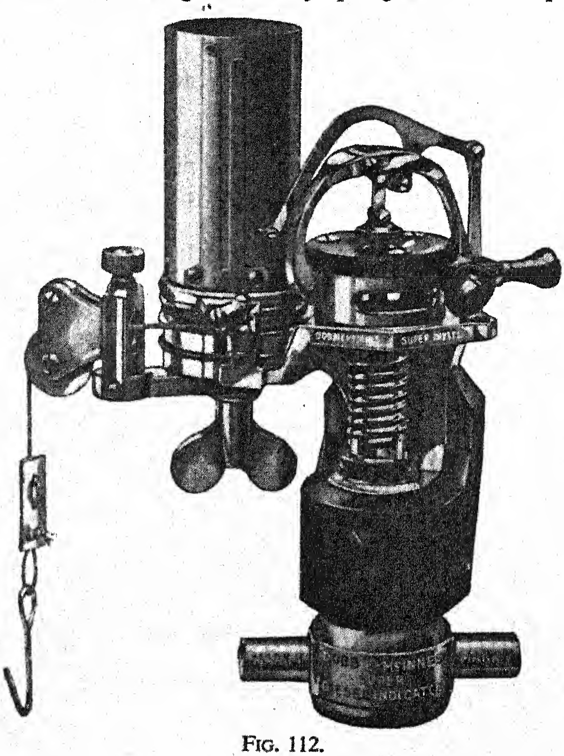


FIG. 112.

for high pressures if the pencil were fixed to the end of the indicator piston rod. Some mechanism is therefore required for magnifying the motion of the indicator piston without distorting this motion, and this mechanism must be very light unless the engine is running at a very slow speed. A number of the older types of indicator give a

very closely accurate enlarged reproduction of this movement, but the mechanism is too heavy for speeds beyond about 120 r.p.m. Any weight attached to a spring has a natural frequency of oscillation of its own, and if this nearly coincides with the engine speed this vibration will

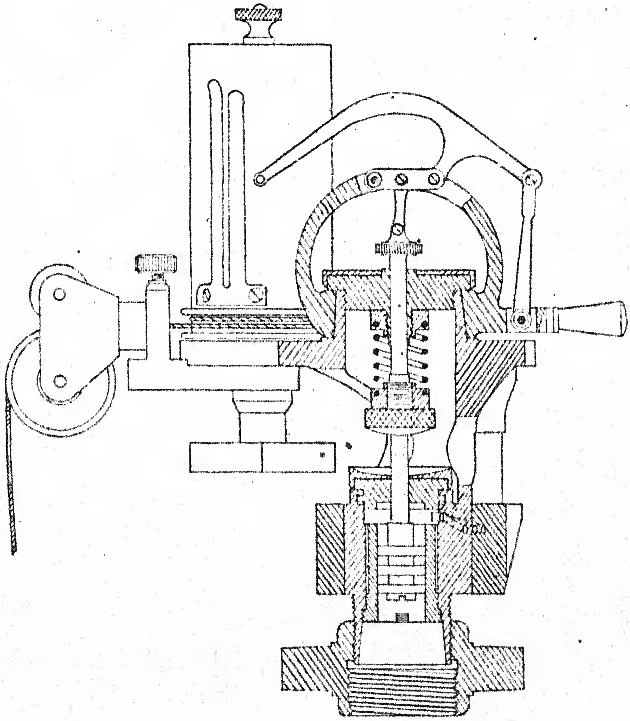


FIG. 113.

be amplified and a wavy diagram will result. The 'natural' frequency of oscillation of the pencil mechanism with the spring used should be very much higher than the engine speed if this kind of diagram is to be avoided, and for this reason the ordinary type of indicator is quite unsuitable for high-speed engines. Special types of indicators are

available for this purpose, but a description of these is beyond the scope of this volume.

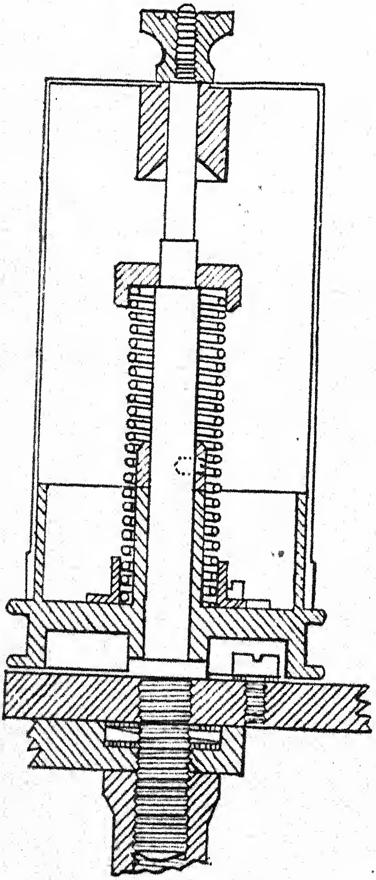


FIG. 114.

The motion of the drum is derived from a 'reducing gear' which is usually actuated by the motion of the cross-head or other suitable part of the engine.

Fig. 112 shows a perspective view and Fig. 113 is a sectional view of the Dobbie McInnes Mark V indicator, specially designed for high pressures and speeds up to 500 r.p.m. It is fitted with an external spring, which is thus unaffected by temperature variations in the cylinder of the indicator, and the pencil mechanism magnifies the motion of the indicator piston six times, at the same time giving the pencil a movement which is, to a high degree of approximation, a straight line parallel to the axis of the drum. The drum is moved forward positively by the cord from the reducing gear against the resistance of the spring shown in Fig. 114 and is returned by this spring.

The Indicator Diagram.—Fig. 116 is an example of a common form of diagram from a single-cylinder non-condensing engine running under good working conditions.

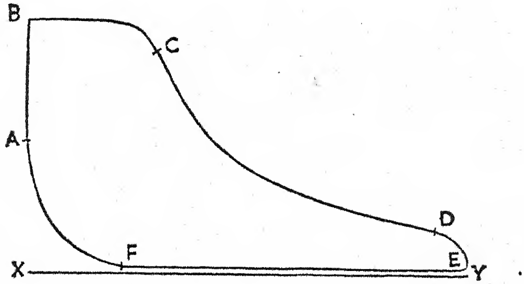
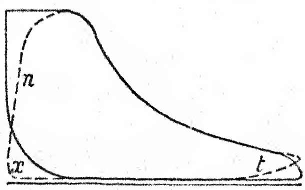


FIG. 115.

The *admission line* AB shows the rise of pressure of the steam as it enters the cylinder.

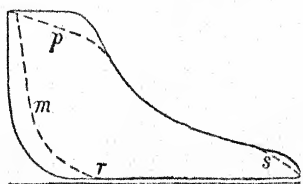
The *steam line* BC shows how nearly the steam pressure in the cylinder reaches that of the boiler.

This difference is obtained by measuring the height of B above the atmospheric line XY with a scale corresponding



n = insufficient lead; *t* = late exhaust; *x* = late compression.

FIG. 116.



m = excess of lead; *p* = wiredrawing; *s* = early release; *r* = early compression.

FIG. 117.

to the scale of the spring in the indicator, and afterwards drawing a horizontal line above B measured with the same scale from XY to represent the pressure in the boiler. Steam will not flow through a pipe unless there is a difference of pressure between the two ends, and the greater the speed of flow the greater will be the drop of pressure. In

addition there is a drop of pressure due to the frictional resistance to flow, so that with a long pipe of too small a diameter the pressure at the delivery end may be appreciably below the boiler pressure. There is always a certain fall of pressure between the boiler and the cylinder in consequence of throttling of the steam in the ports and passages, especially at high speeds, or with too long or too small diameter steam pipes, or with steam pipes having sharp bends.

The effect on the *steam line* of regulating the engine by a throttle valve, and thus varying the opening for the supply of steam, is shown by Fig. 118, which was obtained by successively removing portions of the load on the engine, and maintaining the speed constant by partially closing the steam supply valve.

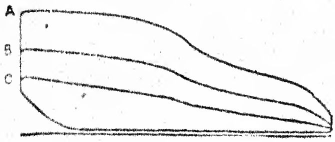


FIG. 118.

The forward pressure line A for a heavy load fell to B for a medium load, and to C for a light load ; the points of cut-off, release, and compression remaining constant.

The *point of cut-off* C, Fig. 115, is a more or less sharp and definite point with trip-gear valves, which cut off suddenly by the action of a strong spring (see Figs. 78 and 81) ; but with the slide valve the cut-off is more gradual, the corner is rounder, and the exact point of cut-off is more difficult to locate (see Fig. 115). In such a case the point of cut-off may be taken at the point where the concave curve of the expansion line meets the convex curve of the cut-off corner.

The effect on the diagram of varying the point of cut-off is shown in Fig. 119 for non-condensing engines, and in Fig. 120 for condensing engines with a trip gear, the cut-off being fairly sharp.

Fig. 121 shows the effect of regulating the power by varying the cut-off in a slide-valve high-speed engine.

In the non-condensing diagram (Fig. 119) with an early

cut-off, it is seen that the expansion line falls below the atmospheric line and forms a loop at the end of the diagram. This is due to the pressure of the steam during expansion falling below atmospheric pressure, and hence, when the exhaust port opens, the pressure will rise, instead of fall, to

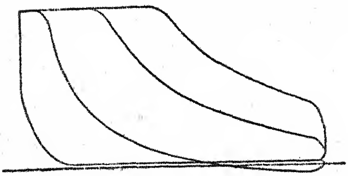


FIG. 119.

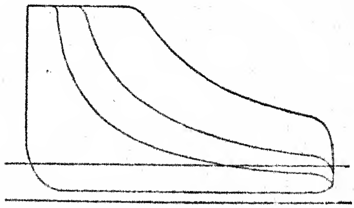


FIG. 120.

the back-pressure line. This is a most wasteful form of diagram.

The *expansion curves* of indicator diagrams vary considerably, and they do not obey any very definite law. They are, in fact, the resultant effect of a variety of separate causes operating to a different extent in different engines, and even in the same engine by change of conditions.

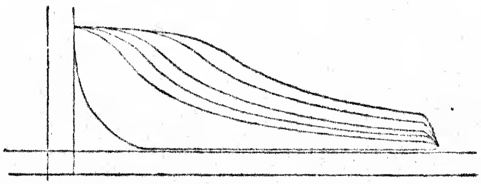


FIG. 121.

The *release point* D (Fig. 115) occurs just before the end of the stroke. With high-speed engines it is important to have an early exhaust, as the trouble is usually not to get the steam *into* the cylinder, but to get it *out*.

The *exhaust line* DE (Fig. 115) represents the fall of pressure which occurs in the cylinder when the exhaust port opens. Fig. 117 shows early opening to exhaust at *s*,

and Fig. 116 shows late opening to exhaust at *t*. A late opening to exhaust, as shown at *t*, is a very grave defect in a diagram.

The *back-pressure line* EF (Fig. 115) shows the amount of the pressure against the piston during its return stroke. With a long exhaust pipe of insufficient size the back pressure in a non-condensing engine may be appreciably above the atmospheric line. In locomotives, for instance, where the exhaust outlet is deliberately restricted in order to obtain a sharp blast, the back pressure is usually well above atmospheric pressure.

The *compression curve* FA (Fig. 115) commences from the point of closure F of the exhaust port. This point depends upon the amount of inside lap on the valve, and the angular advance of the eccentric, and the nature of the curve will depend upon the pressure of the steam trapped, and upon the volume of the clearance space.

CHAPTER XIV

COMPOUND ENGINES

Compound engines are those which have two or more cylinders of successively increasing diameters so arranged that the exhaust steam from the first and smallest cylinder is passed forward to do work in a second, and sometimes a third or fourth cylinder, before escaping to the condenser.

The main object of compounding is to secure a large total ratio of expansion without the disadvantages attached to a very early cut-off in a single cylinder. These disadvantages are :

(1) A large range of temperature, accompanied by excessive condensation and re-evaporation effects unless high superheats are used. The range of temperature is not merely the difference between the initial temperature and the exhaust temperature, but includes the fact that the cylinder is heating up during admission and cooling during expansion and exhaust, so that with an early cut-off the ratio of the cooling period to the heating period is large.

(2) With an ordinary slide valve a cut-off earlier than about 0·3 of the stroke is accompanied by early release and excessive compression. Even when trip gears are used the ordinary type of trip mechanism is not suitable for very early cut-off, say at $\frac{1}{8}$ stroke.

(3) The evil effect of clearance is very much accentuated when the cut-off is very early (see p. 56).

(4) The turning moment on the shaft is more uneven with an early cut-off, so that a greater flywheel effect is required. This effect, however, is reduced if a number of cylinders and cranks are used

In the case of a compound engine a large ratio of expansion may be obtained with a comparatively late

cut-off in each cylinder. The total ratio of expansion is given by

$$R = r_H \times \frac{V_L}{V_H}$$

where R = total ratio of expansion,

r_H = ratio of expansion in the high-pressure cylinder,

V_L = stroke volume of low-pressure cylinder,

V_H = stroke volume of high-pressure cylinder.

Since the stroke is usually the same for each cylinder, the expression becomes

$$R = r_H \cdot \left(\frac{d_L}{d_H}\right)^2$$

As will be seen later, the ratio of expansion in the remaining cylinders after the high-pressure cylinder only affects the *distribution* of work between the cylinders without affecting the total work done, so long as the ratio of expansion in the high-pressure cylinder remains unaltered.

Fig. 122 shows a pair of compound cylinders for a vertical engine. The steam is admitted at A to the high-pressure cylinder. It is exhausted at B, and carried to the low-pressure cylinder through the dotted pipe to the opening C in the low-pressure valve chest. It is exhausted at D to the condenser.

The following diagram (Fig. 123) illustrates the difference between the action of the steam in a simple engine and in a triple-expansion compound engine.

Suppose 1 lb. of steam at 150 lb. pressure absolute admitted to a single cylinder and expanded down to 12 lb. pressure absolute and exhausted into a condenser, when the pressure averages 3 lb. abs. Then the action of the steam in the single cylinder is represented by the whole figure shown cross-lined.

In such a case the temperature in the cylinder would vary from 358° F., the temperature of the steam at 150 lb. pressure down to 142° F., the temperature of the steam at 3 lb. pressure; or a difference of 358−142=216° F. between the initial and final temperature in the cylinder.

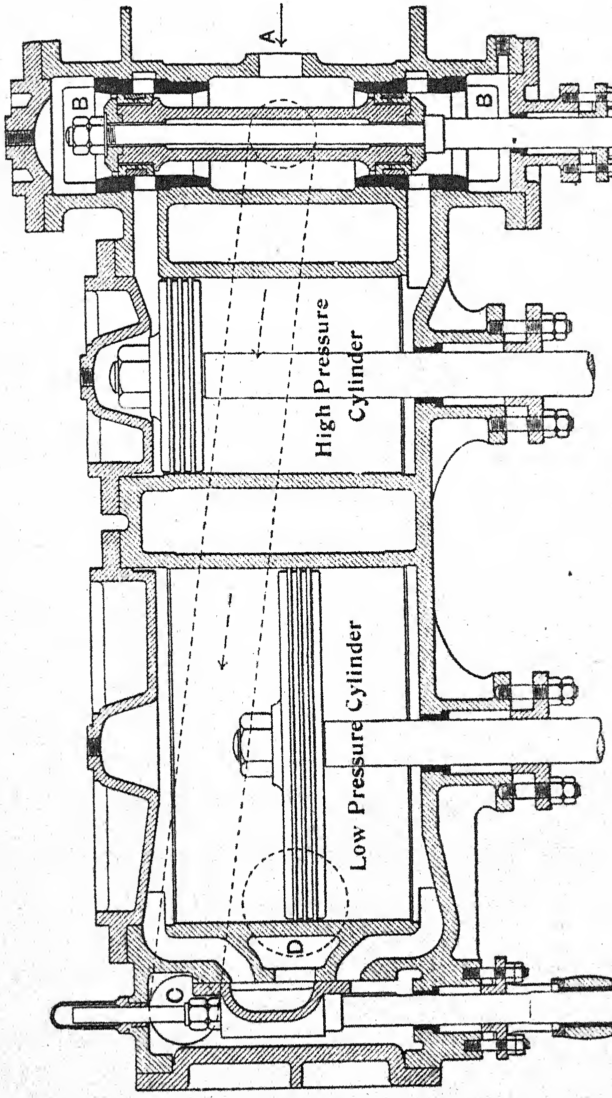


FIG. 122.

And since cylinder condensation increases with the increase in the range of temperature, the loss of steam by initial condensation would here be very great. If now the expansion of the steam be spread over three cylinders (called respectively high, intermediate, and low) the range of temperature in each will be proportionally reduced. Thus in the high-pressure cylinder, working between 150 lb. and

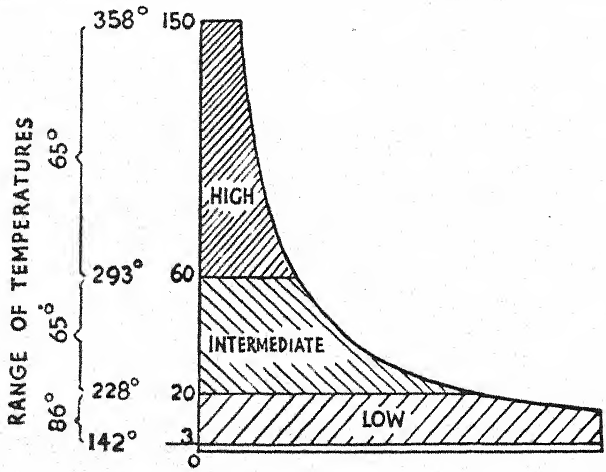


FIG. 123.

60 lb. pressure, there is a variation of 65° F. ; in the intermediate cylinder, working between 60 lb. and 20 lb. pressure, there is again a variation of 65° F. ; in the low-pressure cylinder, working between 20 lb. and 3 lb. pressure, there is a variation of 86° F.

Again, the initial load on the piston of the single-cylinder engine would be equal to forward pressure minus back pressure $=(150-3)\times$area of piston, while the terminal load would be $(12-3)\times$area of piston ; and therefore the initial load is $\frac{147}{9}=16{\cdot}3$ times the terminal load. This would cause a considerable variation of stress on the

working parts, and as the engine must be made strong enough to bear the maximum stresses due to the high initial pressure acting on a large piston area, a much heavier engine would be required than if the stresses were more judiciously distributed. If now the expansion of the steam, the range of temperature, the initial stresses, and the total work are distributed among three cylinders connected with three cranks, a much more economical and mechanically perfect engine is the result.

The shaded parts marked *high*, *intermediate*, *low*, represent the distribution of the work among three separate cylinders.

The diagram further illustrates the historical growth of the steam engine, for the bottom part of the figure represents the condition of the early engines working up to 20 lb. pressure with a single cylinder ; then came higher pressures, higher rates of expansion, and two-cylinder compound engines, and later, with the introduction of steel for boilers, and surface condensation, we have had a rapidly increased boiler pressure and rate of expansion, and the introduction of the three-cylinder or triple expansion compound engine. Pressures are still increasing, while the terminal pressure remains constant, and a fourth cylinder has in many instances been added, forming a quadruple expansion engine.

Figs. 124, 125, and 126 illustrate a two-cylinder compound engine of an older type, HP being the high-pressure and LP the low-pressure cylinder. The steam passes from the boiler by the steam pipe SP into the valve chest of the high-pressure cylinder, where it is admitted to the cylinder and cut off at about one-half or one-third of the stroke ; it is then exhausted by the pipe connecting the two cylinders, shown in Fig. 126 from the high-pressure into the low-pressure cylinder, where it again does work by acting on the low-pressure piston. The steam is then exhausted, either into the air or into a condenser, by a pipe shown below the low-pressure cylinder.

In a two-cylinder compound engine the steam exhausted

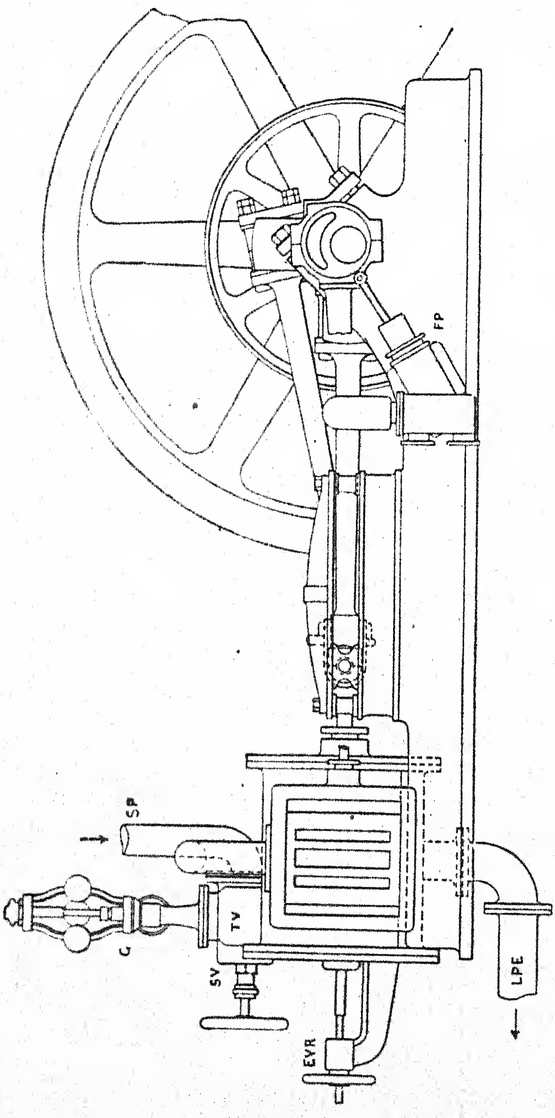


FIG. 124.

HP, high-pressure cylinder ; LP, low-pressure cylinder ; SP, steam pipe ; TV, throttle valve ; SV, stop valve ; G, governor ; LPE, low-pressure exhaust pipe ; FP, feed pump ; EV, expansion valve ; M, main side valve ; EVR, expansion valve regulator ; E, eccentrics ; P, pulley ; FW, flywheel ; CS, crankshaft.

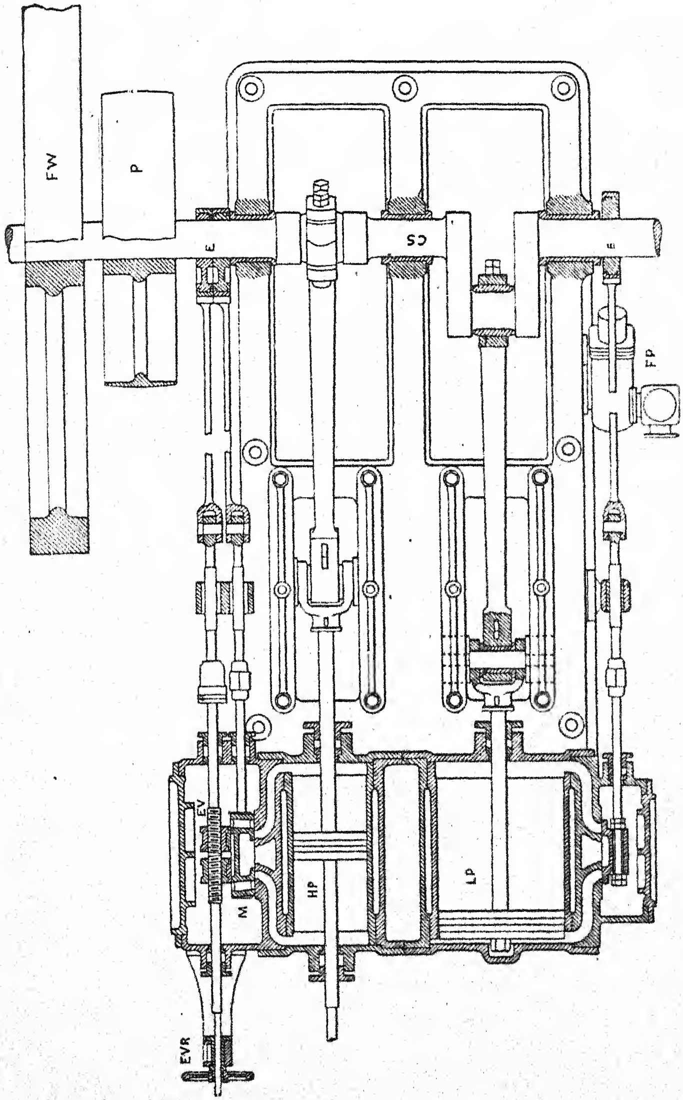


FIG. 125.

from the high-pressure cylinder into the low-pressure acts as forward pressure in the low, and as back pressure in the high, and the effective work done is due to the difference in area between the two pistons.

Thus, suppose steam admitted between two pistons of equal area fixed on a rod, as shown in Fig. 127. Here it is evident that the pressures on the inner faces of the two

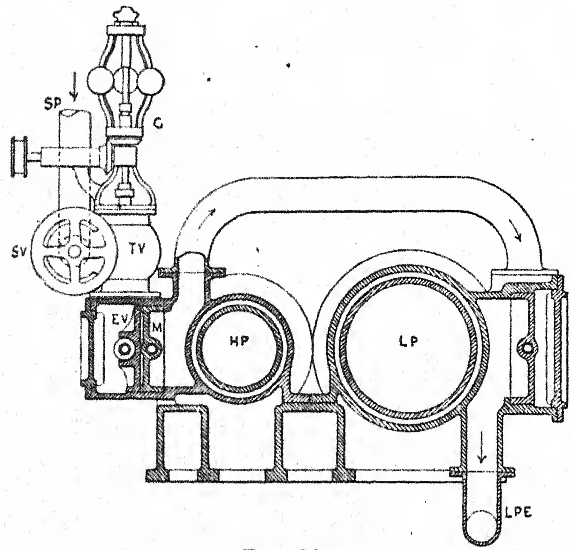


FIG. 126.

pistons being equal and opposite, the pistons will not move in either direction from this cause, and the effective pressure transmitted to the piston rod R by P is quite independent of the pressure between the pistons.

But if the pressure acts on two pistons of *unequal* area, as in Fig. 128, the effective pressure transmitted by the pistons to the piston rod R is equal to the external pressure P on the small piston, plus the internal pressure on the difference of area between the large and small piston, less the back pressure p on the large piston ; from which it

will be seen that the greater the initial pressure P of the steam on the small or high-pressure piston, and the greater the pressure between the two pistons, and the less the back pressure p on the low-pressure piston, the greater the effective pressure transmitted.

The volume of the low-pressure cylinder of a compound

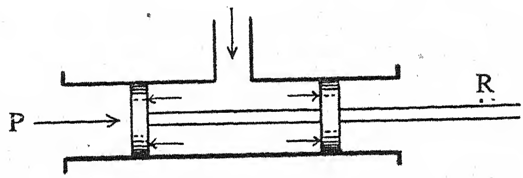


FIG. 127.

engine required for a given power is the same as if the whole of the work to be done, and the whole of the expansions, were performed in that cylinder alone; and its size is therefore estimated as for a single-cylinder engine, to exert the required power with the given initial pressure of

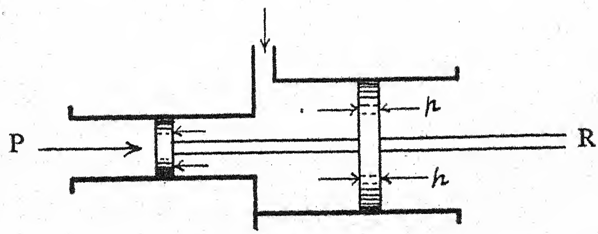


FIG. 128.

steam of the high-pressure cylinder, admitted at once to the low-pressure cylinder and expanded down to the terminal pressure, the assumed point of cut-off being arranged to allow the same number of expansions as with the compound engine.

It will be evident that the volume of steam exhausted

into the condenser at each stroke is the volume due to the capacity of the low-pressure cylinder ; and, provided the terminal pressure is constant, the volume and weight exhausted at each stroke is constant, whether the steam was admitted at boiler pressure direct to the low-pressure cylinder and expanded in it down to the constant terminal pressure, or whether it has arrived there after passing through one, two, or more cylinders.

The size of the low-pressure cylinder having been determined, the remaining cylinder or cylinders are so proportioned as to equalise as much as possible the initial and mean stresses and the range of temperature.

The ratios of the volumes of the cylinders, or of the piston areas (all being of equal stroke), are as the squares of their diameters. Thus, if the low-pressure cylinder diameter be made twice that of the high-pressure, then their areas or volumes are as 1 : 4.

Or, again, if the cylinders of a triple expansion engine have their respective diameters in the proportion of 3, 5, and 8, then the areas of the successive pistons are to one another as $3^2 : 5^2 : 8^2 = 9 : 25 : 64 = 1 : 2{\cdot}78 : 7{\cdot}11$.

The following example illustrates the method of calculating the sizes of cylinders for a multiple-expansion engine.

Example.—Given total ratio of expansion=12, initial pressure=185 lb. per sq. in. abs., back pressure=2 lb. per sq. in. abs., diagram factor=0·8, ratio of cylinder volumes=1 : 3·5 : 7·2, find the necessary diameters of the cylinders of a triple-expansion engine to develop 2500 I.H.P. at 140 r.p.m. The stroke is to be two-thirds of the diameter of the low-pressure cylinder.

$$\text{Mean effective pressure} = 0{\cdot}8 \times \left[185\left(\frac{1+\text{hyp. log } 12}{12}\right)-2\right]$$
$$= 41{\cdot}3 \text{ lb. per sq. in.}$$

Let D = diameter of low-pressure cylinder (in.)

$\frac{2}{3}$D = stroke (in.)

Assuming that all the work is done in low-pressure cylinder

$$\text{Work done per stroke} = 41{\cdot}3 \times 0{\cdot}785D^2 \times \frac{D}{18} \text{ (ft.-lb.)}$$

$$\text{Work done per minute} = 41{\cdot}3 \times 0{\cdot}785D^2 \times \frac{D}{18} \times 280 \text{ (ft.-lb.)}$$

$$\therefore \frac{41\cdot3\times0\cdot785D^2\times D\times280}{18\times33{,}000}=2500$$

$$D^3=163{,}600$$

$$D=54\cdot7 \text{ in. (say 55 in.)}$$

$$\text{Stroke}=36\cdot1 \text{ in. (say 36 in.)}$$

Diameter of high-pressure cylinder $=\sqrt{\frac{1}{7\cdot2}}\times55=20\cdot5$ in.

Diameter of medium-pressure cylinder $=\sqrt{\frac{3\cdot5}{7\cdot2}}\times55=38\cdot3$ in.
(say $38\frac{1}{2}$ in.)

The following sketches show some ways of arranging the steam connections of the cylinders of compound, triple, and quadruple expansion engines :

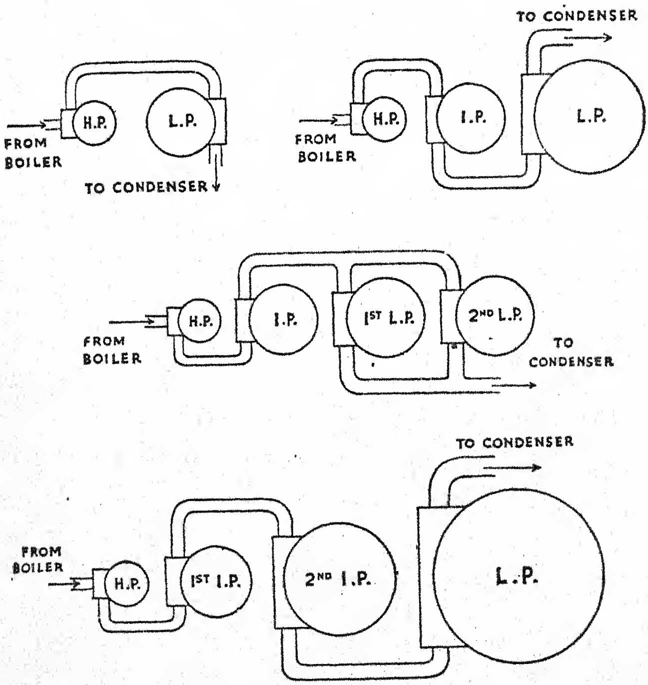


FIG. 129.

The economy of fuel which resulted from the introduction of high-pressure steam, and the compound engine with surface condensation for steamships, was very remarkable, as may be seen from the following table :

Year	Pressure of steam by boiler gauge per sq. in.	Consumption of coal per I.H.P. per hour
1830	2 to 3 lb.	9·0 lb.
1840	8 ,,	5·5 ,,
1850	14 ,,	4·0 ,,
1860	30 ,,	3·0 ,,
1870	40 to 50 ,,	2·6 ,,
1880	70 to 80 ,,	2·2 ,,
1886	150 to 160 ,,	1·5 ,,
1889	—	1·4 ,,
1900	180 to 200 ,,	1·2 ,,

Between 1860 and 1870, when the pressure of steam used for marine engines was about 30 lb. by boiler gauge, and the steam expanded in a single-cylinder, the amount of coal consumed by the best engines was about 4 lb. per I.H.P. per hour.

On the introduction of the compound engine, the consumption fell to a little over 2 lb. per I.H.P. per hour. The triple expansion engine has reduced this to as low as 1·4 lb. per I.H.P. per hour, and the quadruple expansion engine still further reduces the consumption by about 10 per cent. To appreciate the significance of so apparently small a gain as $\frac{1}{4}$ lb. of coal per indicated horsepower per hour, we will take an example :

Suppose a vessel of 6000 I.H.P. steams from London to Melbourne and back in eighty-four days : find the saving on such a trip.

Gain per I.H.P. per hour $=\frac{1}{4}$ lb. of coal.

,, ,, per day $=\frac{1}{4}\times 24$ lb. of coal.

Gain per I.H.P. per 84 days $=\frac{1}{4}\times 24\times 84$ lb. of coal.

Gain per 6000 I.H.P. per 84 days $=\frac{1}{4}\times 24\times 84\times 6000$ lb.

$=3{,}024{,}000$ lb.

$=1350$ tons.

Since 1900 there have been rapid developments in propelling machinery for marine use, including the use of

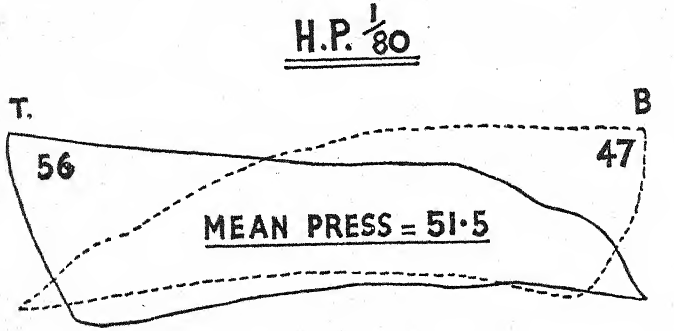
much higher pressures and superheated steam. For large powers steam turbines and gearing are now in use and for the smaller engines (below about 4000 shaft horsepower) exhaust steam turbines are fitted, taking the exhaust from the reciprocating engines at about atmospheric pressure and expanding the steam right down to condenser pressure. Such expansion would be impracticable for pure reciprocating engines, as it would involve low-pressure cylinders of enormous size. The following table illustrates some of the more recent results :

Year	Ship	Machinery	Shaft horse-power	Steam pressure (lb. per sq. in.)	Super-heat (° F.)	Fuel	Fuel consumption (all purposes) in per lb. s.h.p.-hour
1926	*King George*	Turbine	3,500	550	270	Coal	1·08
1928	*Duchess of Bedford*	do.	20,000	375	260	Oil	0·625
1931	*Borneo*	Triple expansion.	3,600	500	280	Coal	1·13 (i.h.p.)
1931	*City of Barcelona*	Triple expansion with exhaust turbine.	4,500	265	—	Coal	1·00

In comparing the figures for fuel consumption with those of the previous table it must be remembered that the figures for coal per indicated horsepower-hour would have to be increased by at least 15 per cent. to obtain the fuel per shaft horsepower-hour for *all* purposes. Note that the *City of Barcelona*, using a lower steam pressure and little superheat, has a much lower fuel consumption than the *Borneo*, using high pressure and high superheat, owing to the increased economy from the addition of an exhaust turbine.

Many ships are now fitted with oil engines for propulsion. The advantages and disadvantages of these are discussed in Chapter XX.

Figs. 130 and 131 show a set of indicator diagrams from a set of marine triple-expansion engines working at full power. The ratios of the cylinder volumes are



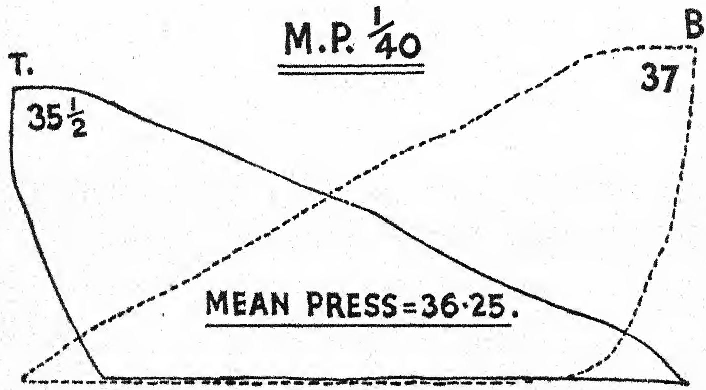
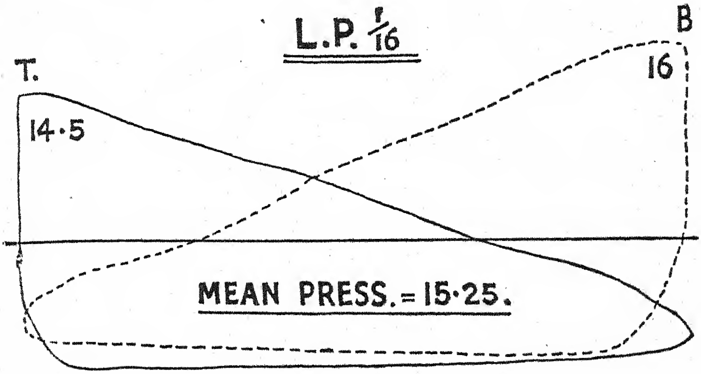


FIG. 130.



PRESSURES.		TEMPERATURES.		I.H.P	
BOILERS	171 LBS	SEA	55° F	H.P.	443
H.P. REC[R]	164 "	DISCH. OVERBD	77° F	M.P.	838
M.P. "	90 "	STEAM	370° F	L.P.	998
L.P. "	15 "			TOTAL	2279
CONDENSER	27"			REVS	68

FIG. 131.

1 : 2·78 : 7·6 ; cut-off in high-pressure cylinder at 0·76 ; revolutions per minute, 68.

The initial pressure in the high-pressure cylinder is shown by the diagram to be 154 lb. per sq. in. Thus it would appear that there is a drop of 171−154=17 lb. per sq. in. between the boiler and engine, probably due to a high speed of flow in the steam pipe and some restriction at the stop valve. The pressure drops during admission to 138 lb. per sq. in. at cut-off. The pressure in the receiver between the high-pressure and medium-pressure cylinders will vary according to whether the receiver is receiving more steam than the medium-pressure cylinder is taking away and vice versa. This is shown by the varying back pressure in the high-pressure and the drop of pressure from admission to cut-off in the medium-pressure cylinder, which is at about 0·7 of the stroke. The admission pressure in the medium-pressure cylinder corresponds closely with the highest back pressure in the high-pressure. The back pressure in the medium-pressure cylinder is fairly constant at 20 lb. per sq. in.

The admission pressure in the low-pressure cylinder is 13 lb. per sq. in. at one end, 17 lb. per sq. in. at the other end, and the admission line droops considerably, showing a somewhat restricted area and volume of receiver pipe for the large volume of steam to be dealt with. The back pressure as shown by the diagram is 11 lb. per sq. in. below atmosphere, which is 2·2 lb. per sq. in. above the pressure in the condenser. This again is accounted for by the high velocity of the steam flowing through the exhaust pipe, which has to pass steam of very large specific volume.

Measurements from the diagram will also show that (1) the ratio of the net loads on the pistons is 1 : 2·38 : 2·33, (2) the temperature ranges in the cylinders are 42°, 81°, and 97° respectively.

CHAPTER XV

BOILERS

The object of a boiler is to generate steam at any desired pressure. Fuel is burned in a furnace of suitable design, some of the heat generated by combustion of the fuel is transferred through metal surfaces to the water in the boiler, and the remainder passes away in the chimney gases or is dissipated by radiation from the external surface of the boiler. Since the fuel used must be paid for, it is important to have as little as possible of the heat generated leaving the boiler unused.

In some cases the type of boiler adopted depends upon the space available or the cost of land or buildings. For instance, marine boilers have usually to be fitted in a small space for the evaporation required, and their weight is generally also a consideration. On the other hand, in the case of a land installation floor area is often expensive in itself and the cost of the building to house the boilers increases with the floor area required.

Boilers may be classified under two main headings: (1) fire-tube boilers, or tank boilers, in which the furnace gases pass *through* tubes of either large or small size, these tubes being surrounded by the water which is to be evaporated; (2) water-tube boilers, in which the gases pass over the external surfaces of tubes through which water is circulated.

Fire-Tube Boilers

The simplest type of boiler, both from the point of view of economy of space and simplicity of construction, is a rectangular box, heated externally by a fire beneath. Some of the earliest boilers, working at a pressure only a few pounds above that of the atmosphere, approximated

to this type. It has, however, some obvious disadvantages. In the first place, much of the heat escapes unused and, in the second place, a flat plate tends to bulge under pressure, and must be stayed. An obvious method of staying flat plates is to have long stays connecting opposite pairs of sides. For instance, the top and bottom plates AB and CD (Fig. 132) would be connected by stays *ss* as shown. Evidently in this case two additional sets of stays would be required for the sides and ends. For low pressures a few stays widely spread would be sufficient, but as the pressure increases more and heavier stays are required until the cost and complication become prohibitive.

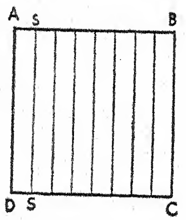


FIG. 132.

Early advantage was taken of the fact that a cylinder under internal pressure retains its shape without any staying, and boilers were made with cylindrical shells and flat ends. Since the flat ends require staying, the cheapest form of construction is to have a shell of comparatively small diameter and correspondingly great length. The Cornish boiler (Fig. 133) and the Lancashire boiler (Fig. 135) are examples of this type. A variation of this is the Vertical boiler (Fig. 144), which occupies very little floor space.

Where (as, for instance, in boilers for marine work) the length must be limited, a boiler of large diameter and small length must be used, and some arrangements for staying the large flat ends are required (see Fig. 145).

In many modern fire-tube boilers the furnace is in a large tube *inside* the boiler and the gases are made to remain in contact with the heating surface for as long as possible in order to waste as little heat as possible. These tubes are under *external* pressure, and must be stiffened by rings or corrugations against collapse.

The Cornish Boiler.—This form of boiler was first adopted by Trevithick, the Cornish engineer, at the time of the introduction of high-pressure steam to the early Cornish engine, and it is still much used.

It consists of a cylindrical shell A, with flat ends, through which passes a smaller tube B containing the furnace, as shown in Fig. 133. The products of combustion pass from the fire grate forward over the brickwork bridge to the end of the furnace tube ; they then return by the two side flues *mm'* to the front end of the boiler, and again pass to the back end of a flue *nn* along the bottom of the boiler to the chimney. Fig. 133 shows a transverse section of the boiler and flues.

One advantage possessed by this type of boiler is that the sediment contained in the water falls to the bottom,

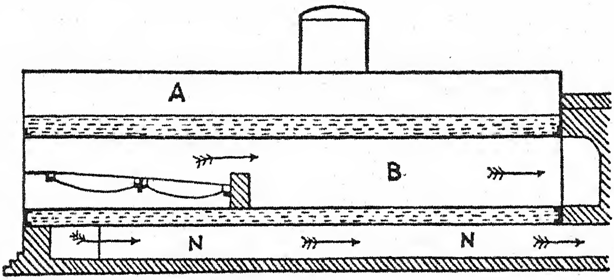


FIG. 133.

where the plates are not brought into contact with the hottest portion of the furnace gases. The reason for carrying the products of combustion first through the side flues, and lastly through the bottom flue, is because the gases, having parted with much of their heat by the time they reach the bottom flue, are less liable to unduly heat the plates in the bottom of the boiler, where sediment may have collected.

Galloway tubes are often fitted to Cornish and Lancashire boilers. Their shape and position will be understood from the diagram, Fig. 134. Holes are cut opposite each other in the furnace tube, and the joints made good by riveting the flanges of the water tube round the hole. Water can thus flow freely through the tube. They pass right across the furnace beyond the furnace bars, so that the flame and hot gases have a considerably increased surface to act upon.

Besides increasing the heating surface, these tubes improve the circulation of the water, and act as a stay to the furnace tube. They are not an unmixed good, however, for they cool the furnace gases and retard combustion.

The Lancashire boiler is very generally employed for stationary boilers. It differs from the Cornish boiler in having two internal furnace tubes instead of one. The separate furnaces are intended to be fired alternately, so that the mixture of smoke and unburnt gases from the newly fired furnace may be consumed in the flues by the aid of the high temperature of the gases from the bright fire of the other furnace.

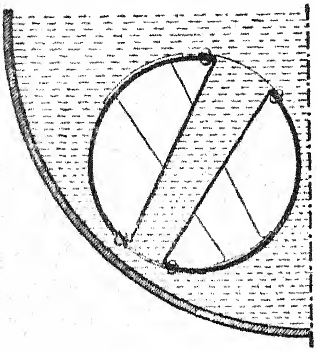


FIG. 134.

Common dimensions of the Lancashire boiler are 7 ft. 6 in. diameter and 28 ft. long, or 8 ft. diameter and 30 ft. long.

The following figures (Figs. 135, 136, and 137) illustrate the construction of a Lancashire boiler. Fig. 135 shows a longitudinal section.

The furnace door, P, opens to the furnace, where the fuel is supported on two or three successive lengths of fire bars, underneath which is the ashpit. At the back end of the furnace is a low brickwork bridge. Besides limiting the length of the fire grate, the bridge helps the complete combustion of the volatile matter from the fuel. The fire bars are supported on bearers. The front bearer, which is a cast-iron plate, is called the *dead plate*. Beyond the furnace are shown the Galloway tubes *aaa*, seen also in Fig. 137. In the Lancashire boiler the furnace gases pass to the end of the furnace tube, and then by the flue underneath the boiler to the front, where it divides and again passes by side flues (see Fig. 137) to the back end of the boiler and

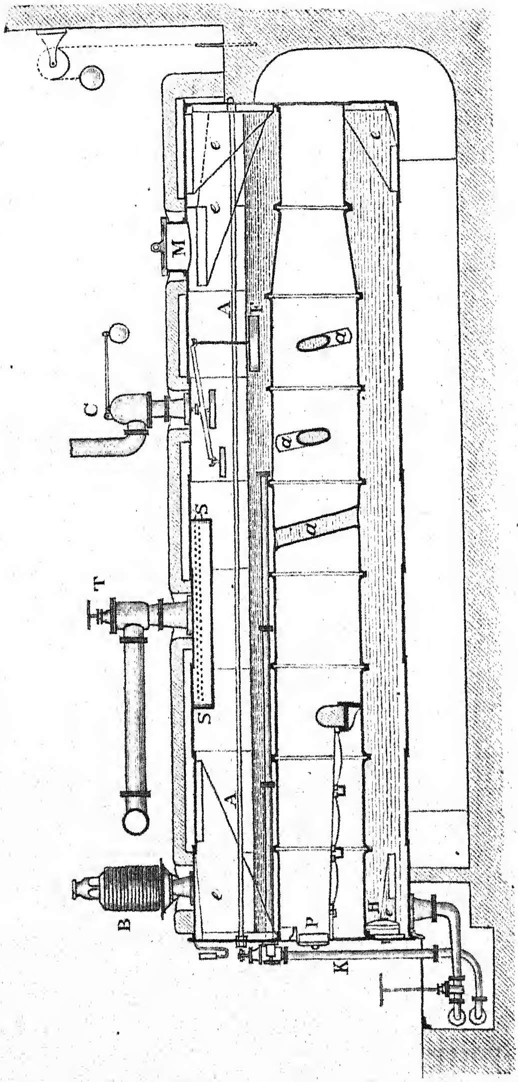


FIG. 135.

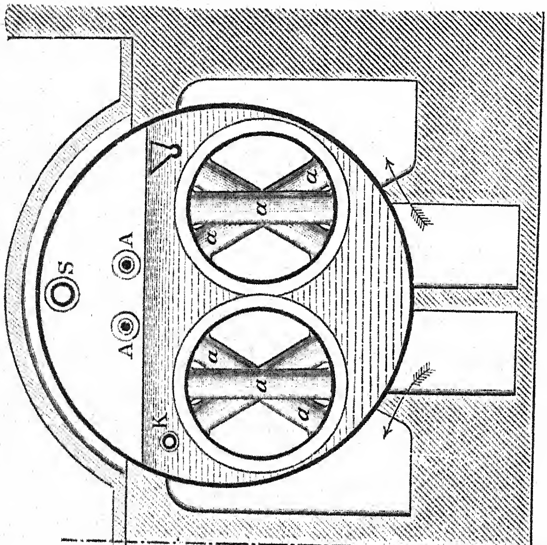


FIG. 137.

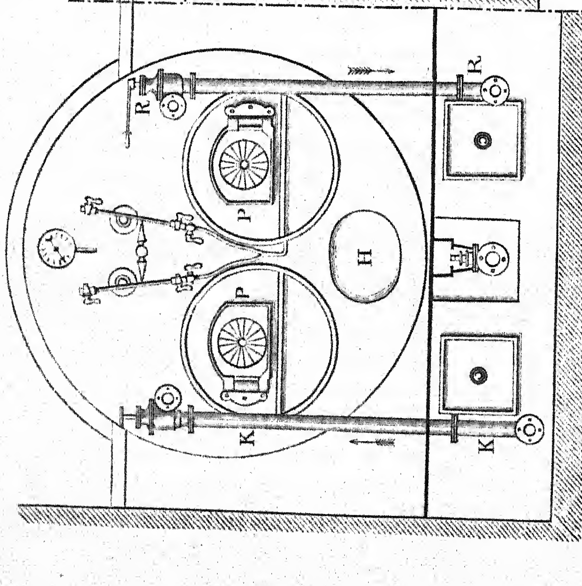


FIG. 136.

up the chimney. The flat ends of the boilers are prevented from bulging by the furnace tubes and by longitudinal stays, AA, also by gusset stays *ee*, shown in Fig. 135 and enlarged in Fig. 138.

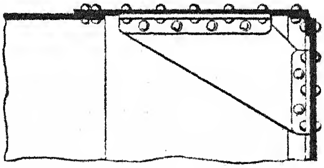


FIG. 138.

The level of the water is shown, and the space above this is occupied by the steam. The steam is conducted from the boiler by the pipe S, which is perforated with holes all along the top so as to admit the steam, and at the same time prevent water-spray from passing to the engine with the steam. On opening the stop valve T (see also Fig. 160), the steam passes by the steam pipe to the engines. Two safety valves are shown, one a *dead-weight* safety valve B (see also Fig. 159), and the other a lever safety valve C.

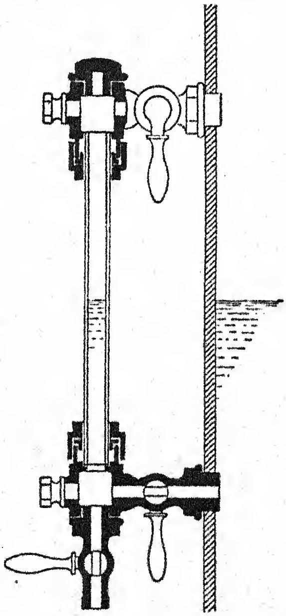


FIG. 139.

The float F is balanced so as to float on the surface of the water. Should the water fall below a safe level, the float F, which falls with the water, causes a small supplementary valve to open by means of levers, and allows steam to escape, giving warning of shortness of water.

A manhole M is shown, by which access is obtained to the interior of the boiler for cleaning and inspection or repairs.

A mudhole H is also required for cleaning out the boiler, and removing the sediment which accumulates.

A blow-off cock and pipe is shown in the bottom of the

boiler at the front end. On the front of the boiler (Fig. 136) is shown a pressure gauge with a finger indicating the pressure of the steam in the boiler above the atmosphere; two water-gauge glasses showing the height of the level of the water in the boiler (see enlarged view, Fig. 139); the furnace doors P; the feed pipe K, which is shown extending some distance into the boiler in Fig. 135; and the scum cock R for blowing off the scum which accumulates on the surface of the water.

The Economiser (Figs. 140, 141, 142).—The greatest source of loss in connection with the boiler is the loss due to the large amount of heat carried away in the flue gases at the chimney. Although this loss is often unnecessarily large, a certain amount of loss is inevitable, owing to the necessity for the gases to be always hotter (by say 100°) than the water in the boiler.

If, however, a series of pipes be placed in the flues beyond the boiler, and the feed water be passed through them on its way to the boiler, much of the heat from the gases may be saved and restored to the boiler in the feed water. In this way a saving of from 10 to 15 per cent. may be realised.

Green's Economiser consists of a nest of cast-iron pipes (Fig. 140) 4 in. diameter and 10 ft. long. The feed water is pumped through these pipes in a direction opposite to that of the flue gases. In order to keep the outer surfaces of the tubes clean, a system of scrapers is adopted, driven automatically and continuously by a chain gear, the scrapers travelling up and down the tubes slowly by means of a belt-driven gear, the belt being reversed automatically in much the same way as the table of a planing-machine is reversed. A safety valve and blow-off valve are also fitted.

Fig. 142 shows how the flue gases may be bye-passed by means of dampers so as to pass through the economiser on their way to the chimney, or direct to the chimney without passing through the economiser.

Economisers, being more or less of an obstruction in the flue, as well as cooling the chimney gases, always decrease the draught to some extent, and they should not be fitted

where the draught is already defective, unless some mechanical means of increasing the draught is also added.

Economiser heating surface is cheaper than boiler heating

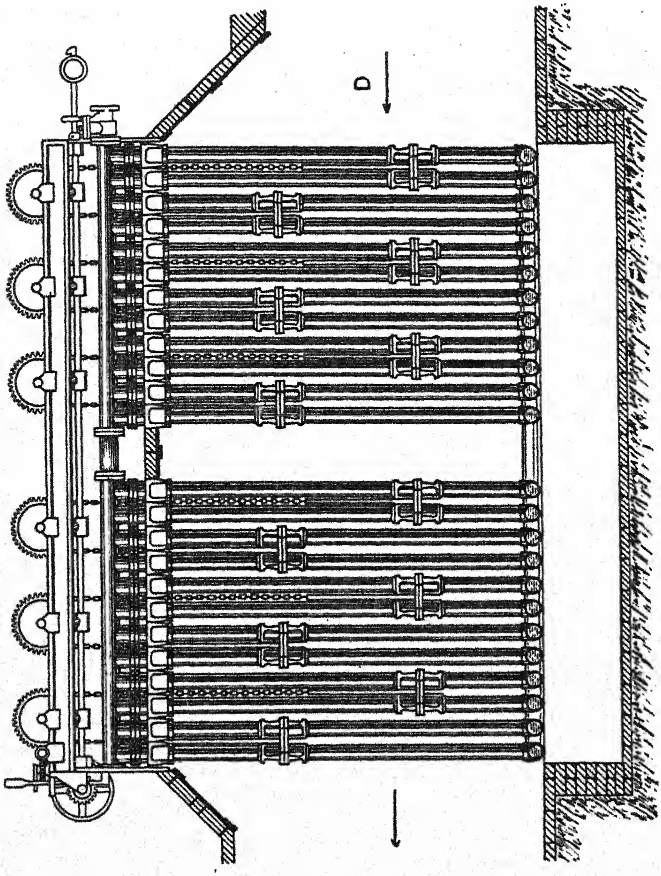


FIG. 140.

surface, and it is also more efficient than the later portion of the boiler heating surface, because the difference of temperature between the flue gases and the feed water in

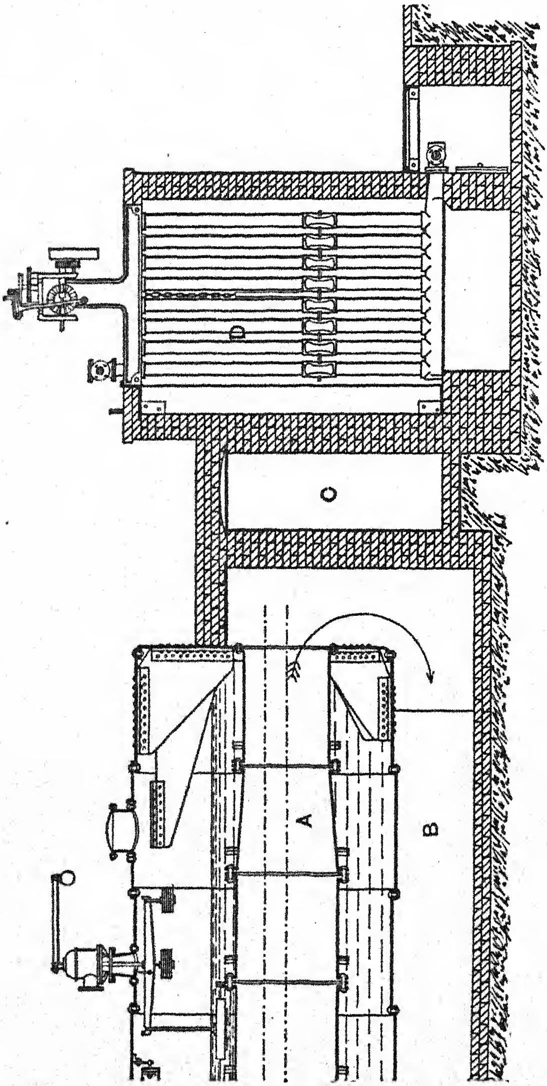


FIG. 141.—A, boiler flue ; B, flue under boiler ; C, receiving chamber from side flues ; D, economiser.

the economiser is greater than the difference of temperature between the flue gases and the water in the boiler.

Thus if the flue gases on leaving the boiler are at a temperature of 600° F., and the temperature of the steam and water in the boiler is 350° (Fig. 143), then the tempera-

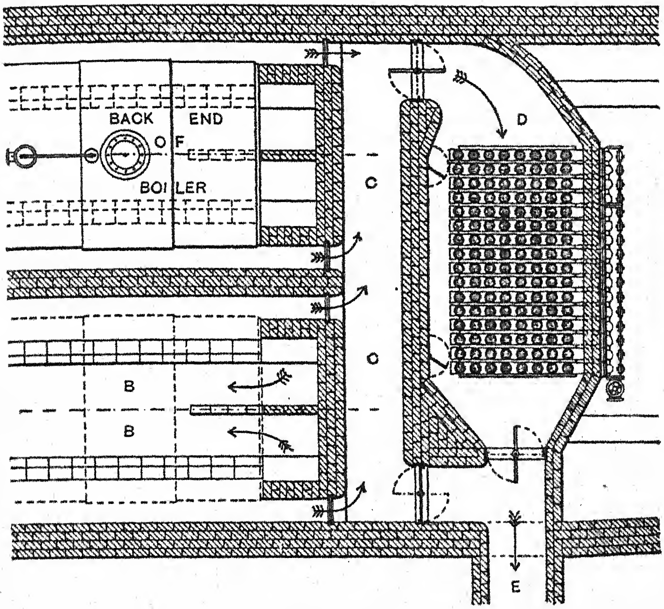


FIG. 142.

ture difference at the later portion of the boiler surface is—

$$600-350=250° \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Again, if the temperature of the flue gases entering the economiser is 600° F., and on leaving it is 400° F., then the mean temperature of the gases is $(600+400)\div2=500°$ F. And if the temperature of the feed water entering the economiser is 100° F., and on leaving it is 240° F., then the mean temperature of the water in the economiser is $(240+100)\div2=170°$ F.

The mean temperature difference on the two sides of the economiser heating surface will therefore be—

$$500-170=330° \text{ F.} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If the feed water is admitted to the economiser tubes quite cold, the cold pipes act as a condenser to the moisture in the flue gases, some of which is deposited on the outside of the colder tubes, and in consequence external corrosion of the tubes takes place. This corrosion is accelerated by

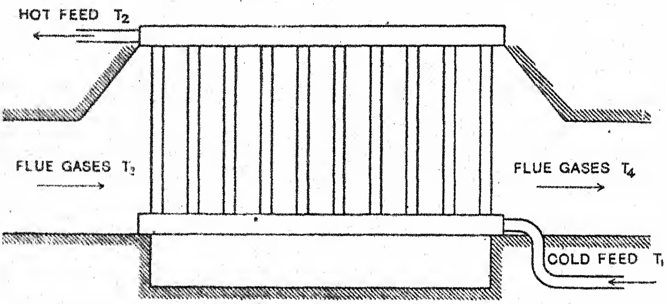


FIG. 143.

the fact that the gases contain a certain amount of SO_2, which in the presence of water is very corrosive.

To prevent this action, the feed water, before being admitted to the economiser, is heated by a small jet of steam.

Percentage Gain by the Use of an Economiser.—Suppose the steam and water in the boiler to be at 350° F. (or 120 lb. pressure by gauge). The total heat of the steam at that pressure is 1197 units reckoned from 32° F., or if the water is supplied at 100° F., then $1197-(100-32)=1129$ units reckoned from 100° F.

Now, if the feed water entered the economiser at 100° F. and left it at 240° F., we have a gain of $(240-100)=140$ units of heat per pound;

$$\text{or}=\frac{140\times100}{1129}=12{\cdot}4 \text{ per cent.}$$

Vertical Boilers

The illustration, Fig. 144, shows the construction of a vertical boiler. These boilers are used for small powers, and where space is limited. The internal fire box is frequently made slightly tapering towards the top, to allow of the ready passage of the steam to the surface. The bottom of the fire box is attached to the bottom of the outer shell

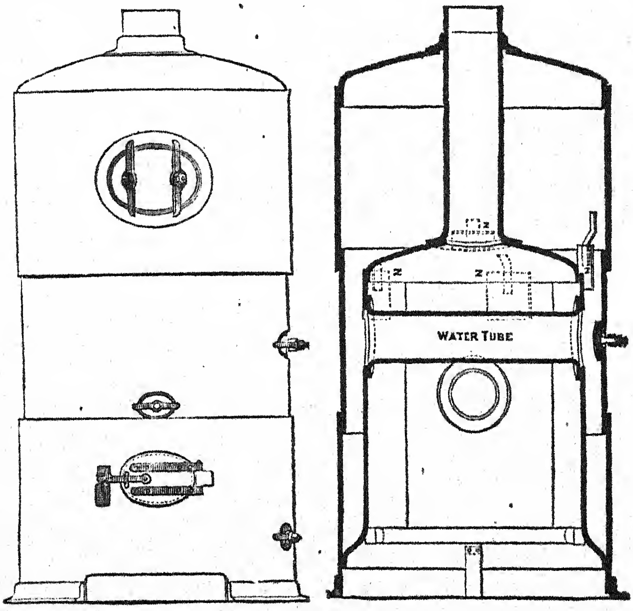


Fig. 144.

by being flanged out as shown, or by means of a solid wrought-iron ring, as shown in the locomotive boiler, Fig. 148, the rivets passing right through the plates and solid ring. The water tubes pass across the internal fire box, and increase the heating surface as well as improve the circulation, though they cool the furnace gases. The plate forming the passage leading from the top of the fire

box to the chimney—called the *uptake*—is frequently protected either with fireclay or with a cast-iron liner.

Owing to the short path from the furnace to the chimney these boilers have a low efficiency. Other vertical types, in which the gases pass over or through a nest of tubes in the boiler, have a higher efficiency.

The Marine Boiler

Marine boilers of the 'Scotch' or tank type are still used for many merchant steamers. Fig. 145 shows a boiler of the tank type, constructed to carry steam at pressures up to 150 or 160 lb. per sq. in.

Description of the Figure.—The boiler is of the cylindrical, multitubular type, fired from one end, with three furnaces. The products of combustion in the furnaces are carried forward by the draught into the combustion chambers CC, and thence through the tubes in the direction of the arrow to the front of the boiler, whence they pass up the funnel.

The *outside shell* is 12 ft. $1\frac{5}{8}$ in. extreme diameter, and 9 ft. $5\frac{13}{16}$ in. extreme length. The plates are of mild steel, $\frac{13}{16}$ in. thick, in three rings united together circumferentially by double-riveted lap joints. The longitudinal seams are treble riveted. The end plates are made in three pieces, and are joined together by double-riveted lap joints, and flanged to meet the shell and the furnace flues.

The *furnaces* are 3 ft. inside diameter, constructed of Fox's corrugated steel plates $\frac{1}{2}$ in. thick. They are flanged at the back end, and riveted to the combustion chambers.

The *combustion chambers* are flat on the top and are supported by wrought-iron girder stays. The back and sides of these chambers are stayed with $1\frac{3}{8}$-in. screwed stays, fitted with nuts on both ends.

The boiler contains 200 steel tubes, 3 in. diameter outside, of which 42 are stay tubes. The stay tubes are of wrought iron, $\frac{5}{16}$ in. thick, and screwed into the plates with nuts on the front ends.

Longitudinal stays, $1\frac{7}{8}$ in. diameter, steel, pass through

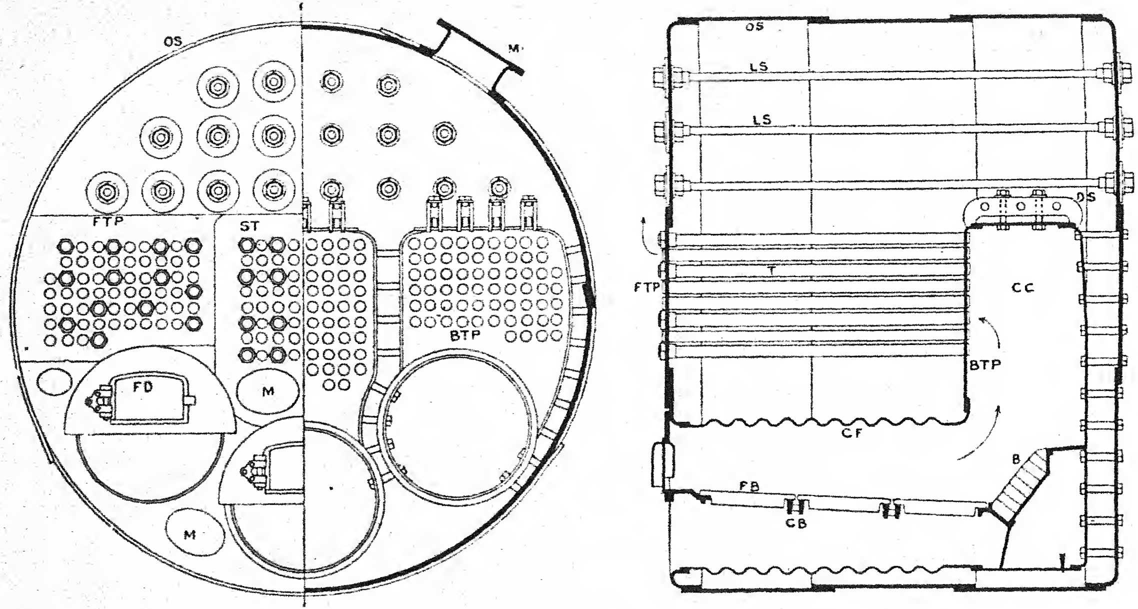


FIG. 145.

OS, outer shell ; CF, corrugated flues ; FB, fire bars ; B, brickwork bridge ; CC, combustion chamber ; BTP, back tube plate ; **FTP**, front tube plate ; LS, longitudinal stays ; DS, dog stays ; ST, stay tubes ; FD, fire door ; M, manhole ; CB, cast-iron bearer.

the steam space from end to end, and support the front and back plates of shell.

Fire-grate area=area of grate × number of furnaces
$=(3\times 6)\times 3=54$ sq. ft.

This type of boiler has changed but little in form during past years, though it has greatly increased in strength in order to enable it to meet the steadily continued demand for increase of boiler pressure.

The object of providing a large number of tubes inside the boiler, instead of a single large tube, is to obtain a larger heating surface with the same total sectional area. For instance, a tube 20 in. diameter and 100 in. long will give a sectional area of 314 sq. in. and a total surface area of 6280 sq. in. If we replace this with 100 tubes of 2 in. diameter, the total sectional area is $3{\cdot}14\times 100=314$ sq. in., as before, but the surface area is now $100\times 6{\cdot}28\times 100$ $=62{,}800$ sq. in., which is ten times as much.

Fig. 146 represents a double-ended marine return tube boiler. The gases pass from the furnace into the combustion chamber, and then return in the opposite direction through the small tubes to the front of the boiler, whence they pass up the funnel.

The Heating Surface.—The effective heating surface of a marine boiler is obtained by finding the sum of the following areas :

1. Area of furnace above level of fire bars.
2. Area of sides and crown of combustion chamber above level of bridge.
3. Area of back tube plate, less area of holes for tubes.
4. Area of surface of tubes, namely, the area obtained by multiplying the internal circumference by the length between the tube plates. The area of the front tube plate is omitted.

The length of furnace should not exceed 6 ft., otherwise it becomes difficult to stoke. The fire doors are made of three pieces of plate placed about $2\frac{1}{2}$ in. apart, the two inner ones being perforated. It will be noticed (Fig. 145) that the back of the combustion chamber slopes a little

inwards towards the top. This enables the steam to rise more freely.

The space allowed between the tubes is 1 in., and the tubes are arranged in vertical rows to allow of the boiler being properly cleaned internally.

Manholes are placed on the top and front of the boiler, to get at the upper and lower parts of the furnaces for cleaning and repairing. The furnace bars are of wrought iron, and in three lengths, sloping towards the bridge $\frac{3}{4}$ in.

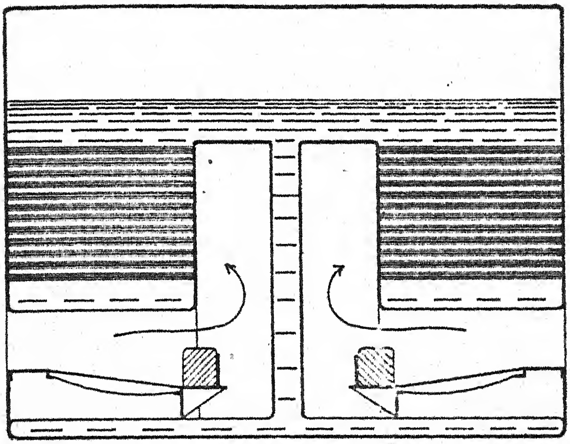


FIG. 146.

per foot. Distance between bars $\frac{1}{2}$ in., maintained by widened ends of bars.

Steam Room.—It is important to have as large a reservoir of steam as possible above the level of the water in the boiler, to prevent too great fluctuations of pressure. The water level should be at least 7 in. above the top row of tubes.

To find the cubic contents of the steam space : Find the area of the segment of the circle occupied by the steam, and multiply by the internal length of the boiler.

To find the area of the segment of a circle (Fig. 147):

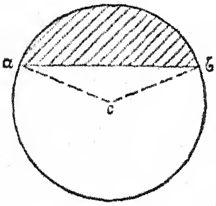


FIG. 147.

$$\text{Area of whole circle} \times \frac{\text{angle } acb}{360} -$$

area of triangle *abc*.

To give the front and back plates of shell the necessary stiffness, large circular plate washers, 10 in. diameter, are riveted on to outside of plates.

The maximum stress allowed on these stays is 8000 lb. per sq. in. for stays under $1\frac{1}{2}$ in. diameter, and 9000 lb. for stays over $1\frac{1}{2}$ in.

THE LOCOMOTIVE BOILER

The following diagram (Fig. 148) is a longitudinal section of a locomotive boiler. The fire box FB, or furnace, is of rectangular section, and is made of copper, stayed by means of screwed and riveted copper stays, $\frac{7}{8}$ in. in diameter and 4 in. apart, to the outer shell of the boiler.

The crown plate of the fire box being flat requires to be very efficiently stayed, and for this purpose girder stays called fire-box roof stays are mostly used, as shown in the figure. These stays are now being made of cast steel for locomotives. They rest at the two ends on the vertical plates of the fire box, and sustain the pressure on the fire-box crown by a series of bolts passing through the plate and girder stay, secured by nuts and washers. Fig. 149 is a plan and elevation of a wrought-iron roof stay.

Another method adopted in locomotive types of marine boilers for staying the flat crown of the fire box to the circular shell plate is shown in Fig. 150—namely, by wrought-iron vertical bar stays secured by nuts and washers to the fire box and with a fork end and pin to angle-iron pieces riveted to the outer shell.

The barrel of the boiler contains the tubes through which the products of combustion pass. The advantage of the tubes is the large amount of heating surface they expose to

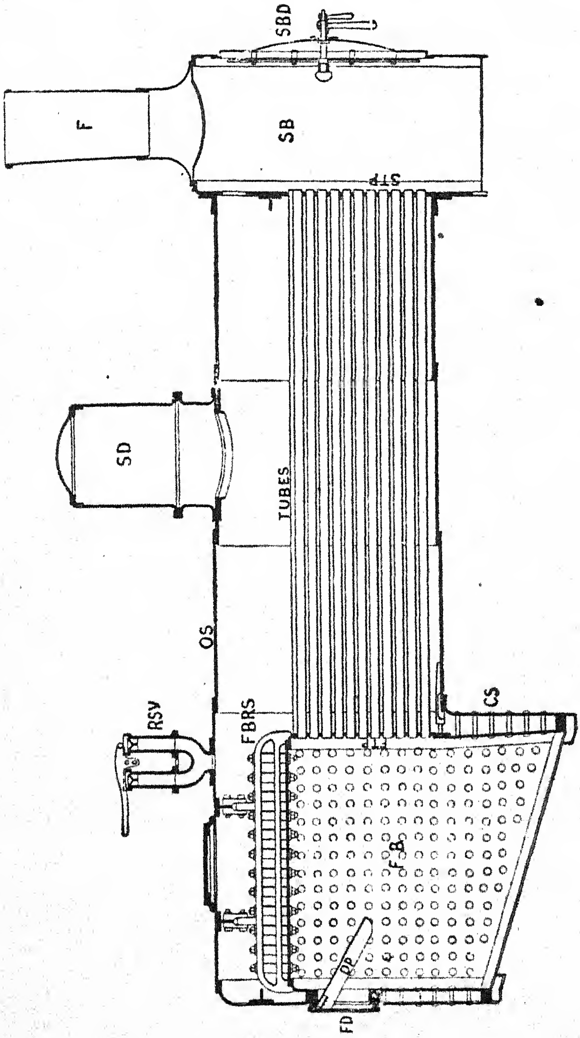


FIG. 148.

FB, fire box; FD, fire door; DP, deflector plate; FTP, fire-box tube plate; FBRS, fire-box roof stays; STP, smoke-box tube plate; SB, smoke box; SBD, smoke-box door; SD, steam dome; OS, outer shell; RSV, Ramsbottom safety valve; F, funnel or chimney.

the heated gases. If the tubes are placed too close together the steam generated round the tubes cannot freely escape ; and as steam cannot absorb the heat so readily as water, the surface of the tube is liable to be overheated and to deteriorate rapidly. The part of the tube nearest the

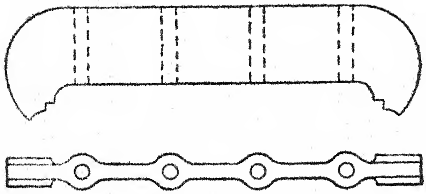


FIG. 149.

fire box is the most effective heating surface ; and the value of the heating surface of the tube rapidly decreases towards the smoke-box end.

The upper surface of the tube is also far more effective than the lower, even when the tube is clean ; but when soot is deposited in the lower portion of the tube, that part of it is valueless as heating surface.

The chamber beyond the tubes and below the chimney is called the smoke box, SB. A dome, SD, is sometimes pro-

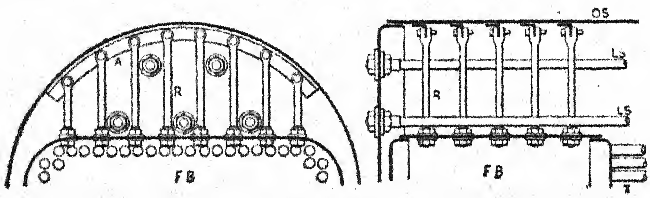


FIG. 150.

vided, from which the steam is taken to supply the engines ; and a safety valve, RSV, is placed as shown.

WATER TUBE BOILERS

There are two main divisions which may be made in these boilers, namely, the large-tube type and the small-tube type. In the large-tube type, the internal diameter of the tubes is $3\frac{1}{2}$ to 4 in. ; in the small-tube type, the internal diameter is about 1 in., thickness of tube from $\frac{1}{10}$ to $\frac{1}{8}$ in.

Water-tube boilers have the following advantages :

(1) For a given strength and a given internal pressure, the thickness of a cylinder should be proportional to its diameter. Thus if the shell of a boiler 7 ft. in diameter under an internal pressure of 300 lb. per sq. in. needs to be $1\frac{1}{4}$ in. thick, then for the same strength a tube 4 in. diameter need only be $\frac{4}{84}\times1\frac{1}{4}=0{\cdot}06$ in. thick. Since a tube of this size would be made at least $\frac{1}{8}$ in. thick, it would be at least twice as strong as the larger cylinder or, to put it in another way, if it corroded to half its thickness it would still be safe for the pressure stated. A further advantage of the boiler composed of comparatively small tubes as compared with the 'tank' boiler is that the bursting of the shell of a tank boiler usually involves considerable damage to property and risk of loss of life, whereas if a tube bursts in a water-tube boiler the steam passes away to the flues unless the fire door happens to be open at the time, and even in this case the risk to life is much smaller. The boiler, of course, is shut down for the time, but the tube ends can be plugged temporarily and the boiler got to work again in quite a short time if necessary.

(2) A considerable amount of heating surface can be packed into a small space. This is of great importance in the case of power stations with very large outputs and in naval vessels and large liners where the available space for boilers is restricted.

(3) Owing to the very large heating surface and good circulation, also freedom for expansion and contraction, steam can be raised very quickly from cold and rapid fluctuations of load can be dealt with.

(4) The hot gases travel approximately at right angles to the tube surface and so give up heat more rapidly than if they are moving parallel to the tube surface.

(5) Ease with which they can be transported. The parts can be transported separately and the boiler built up on the site.

Small tube boilers are only used when very high rates of evaporation are required in a confined space. The tubes,

being thin, have a shorter life and the cost of repairs is usually high.

The main disadvantages of water-tube boilers are :

(1) If the water contains scale-forming materials, a small deposit of scale will lead to overheating and burst tubes.

(2) When the tubes are nearly horizontal, a deposit of soot and ash tends to form on the upper surfaces of the tubes, reducing their capacity for transmitting heat. 'Soot blowers' must be used fairly frequently.

(3) The quantity of water carried is small compared with the tank type, so that the water level must be watched carefully. Feed-water regulators are usually fitted, which accelerate or slow down the feed pump as the water level falls or rises.

(4) The cost of upkeep is relatively high.

The Babcock and Wilcox Water-tube Boiler.—This boiler, illustrated in Figs. 151 and 152, is composed of weldless mild steel tubes from 3 to 4 in. diameter, placed in an inclined position and connected with each other and with a horizontal steam and water drum by vertical passages at each end. A mud drum is attached to the rear and lowest point of the boiler in which sediment collects, and from which it may be blown off.

The end connections are in one piece for each vertical row of tubes. The openings for cleaning at the ends of each tube are closed by plates ; the joints are carefully machined, and the plates are held in place by wrought-iron clamps and bolts.

Above the tubes is the horizontal steam and water drum, the water level being kept at about the middle of the drum, the remainder being steam space.

In Figs. 151 and 152 additional sets of tubes of U-shape, fixed horizontally, are fitted in the chamber between the water tubes and the drum for the purpose of *superheating* the steam. The steam passes from the steam space of the drum through the perforated pipe shown in the steam space ; it then passes downwards into the superheater, entering the superheater at the upper part of the bend and

leaving it at the lower part, from whence it is carried by a pipe to the stop valve, and delivered thence to the engine.

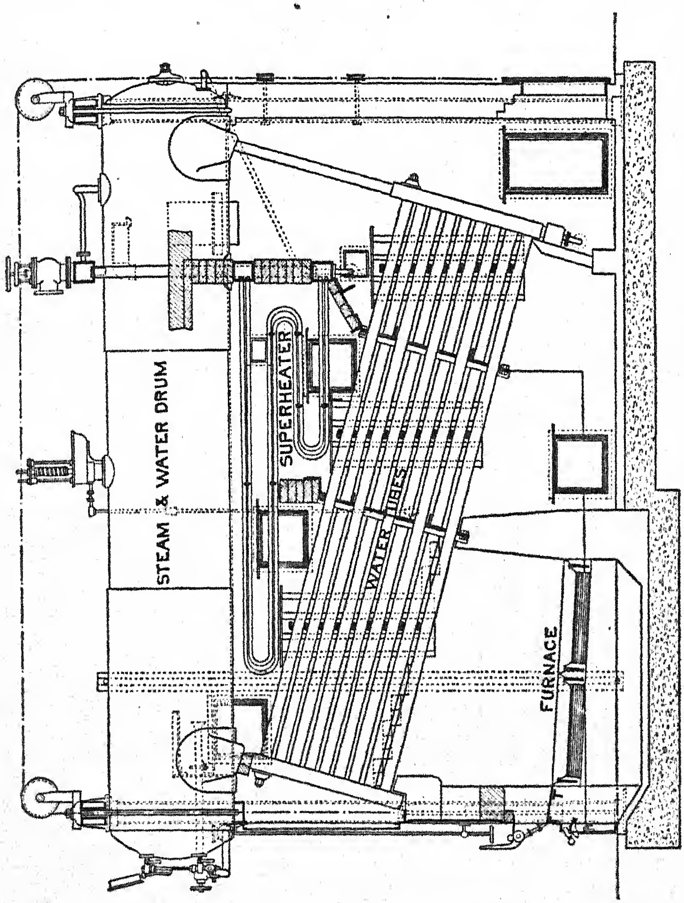


FIG. 151.—Babcock and Wilcox Boiler.

The Yarrow Boiler (Fig. 153).—This boiler consists of an upper cylindrical steam and water drum, into the lower portion of which are inserted two sets of straight tubes

spread at the bottom so as to make an angle of 60° with each other, and to form a furnace space between the two sets of tubes. The bottom ends of the straight tubes are fixed into a flat tube plate, which is covered by a semicircular cover, forming a water chamber at the lower end of each set of tubes.

The boiler is enclosed in a sheet-iron casing, and outside

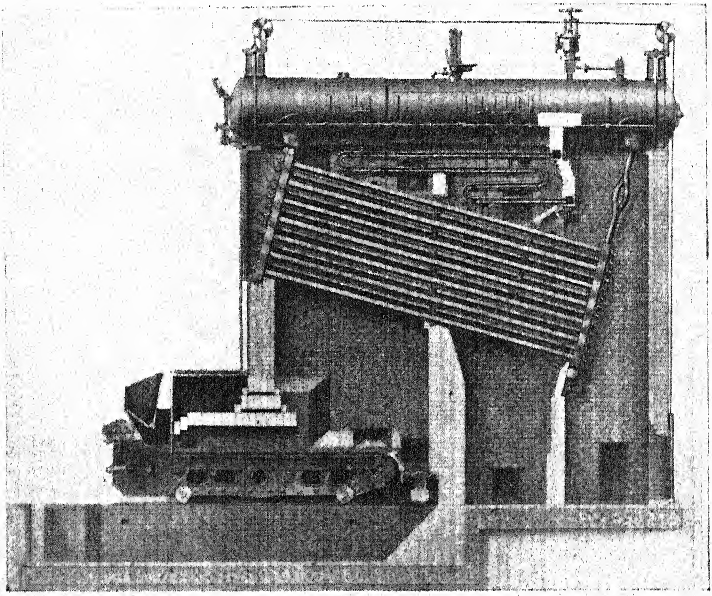


FIG. 152.

the whole a further casing to prevent loss of radiated heat. The furnace gases pass among the tubes on their way to the funnel.

The ends of the tubes of the Yarrow boiler deliver their contents below the water-level in the upper drum. The circulation of the water arranges itself in the tubes, the currents flowing upward through the inner hottest tubes, and downward through the outer cooler tubes.

The frontispiece shows a vertical section of a Stirling tri-drum boiler with 'inverted' type superheater, chain grate stoker and economiser. The capacity is 40,000 lb. normal with an overload of 50,000 lb. per hour at a pressure of 315 lb. per sq. in., while the designed steam temperature

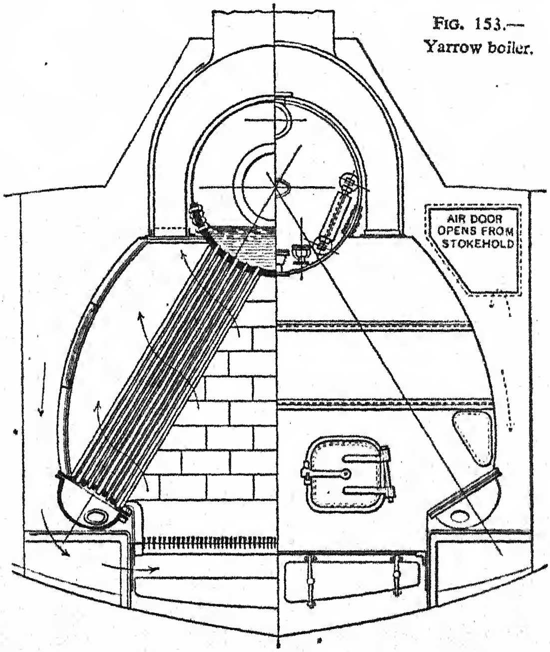


FIG. 153.—Yarrow boiler.

600° F. The heating surface is 6529 sq. ft. From the vertical section of the boiler, showing the furnace, superheater, economiser and forced and induced draught fans, it will be seen that a Babcock and Wilcox chain grate stoker is installed. This has a grate area of 162 sq. ft., being 18 ft. long and 9 ft. wide. Water boxes are arranged immediately above the grate level in order to prevent the formation of clinker. The stoker is motor driven through a variable speed gear giving 4 speeds.

Liptak suspended arches are provided at the front and rear of the stoker, that at the front having a very short

ignition arch and sweeping upwards at an acute angle immediately after the ignition zone.

It will be seen that the superheater is of the 'inverted' type, with the headers at the bottom, thus being self-draining.

At the back of the boiler is a gilled tube economiser consisting of 161 cast iron horizontal gilled tubes 11 ft. in length, the total heating surface being 4936 sq. ft.

The forced draught fan is situated in the basement and the induced draught fan is situated on the firing floor at the back of the boiler, the arrangement of fans and dust collector being shown in the illustration. Steam and water connecting tubes are provided between the upper drums.

The hot gases from the furnace are suitably directed through the banks of tubes by means of firebrick baffle tiles arranged at the back of the banks of tubes as shown.

This boiler typifies the bent-tube class of water-tube boiler, and it is claimed for the curved tubes that smaller strains are generated in the boiler due to varying temperatures. Other advantages of nearly vertical tubes are that accumulations of soot and dust are less likely than with horizontal tubes and that scale and internal deposit tend to fall downward into the lower drum, which is not exposed to the highest temperatures and from which they can be removed without difficulty.

Fig. 154 shows the lay-out of a double bank of boilers. for a large power station. These boilers are equipped with chain grate mechanical stokers, superheaters, economisers, and air heaters. Each boiler is capable of evaporating 120,000 lb. of feed water per hour to steam at 375 lb. per sq. in. and 730° F., the efficiency at full load being 84 per cent. The path of the gases from the furnace to the flues is shown by arrows. The grate area is 512 sq. ft., boiler heating surface 16,000 sq. ft., superheater surface 6000 sq. ft., economiser surface 12,000 sq. ft., air heater surface 12,000 sq. ft. The height from firing floor to top of air heater is 45 ft.

SUPERHEATERS

The steam produced in a boiler in the presence of water is called saturated steam, and may be either 'dry' or

' wet ' steam. The temperature of *saturated* steam depends upon the pressure, and may be obtained from the Steam Tables. By the application of additional heat to saturated steam after it leaves the boiler, the steam becomes *super-*

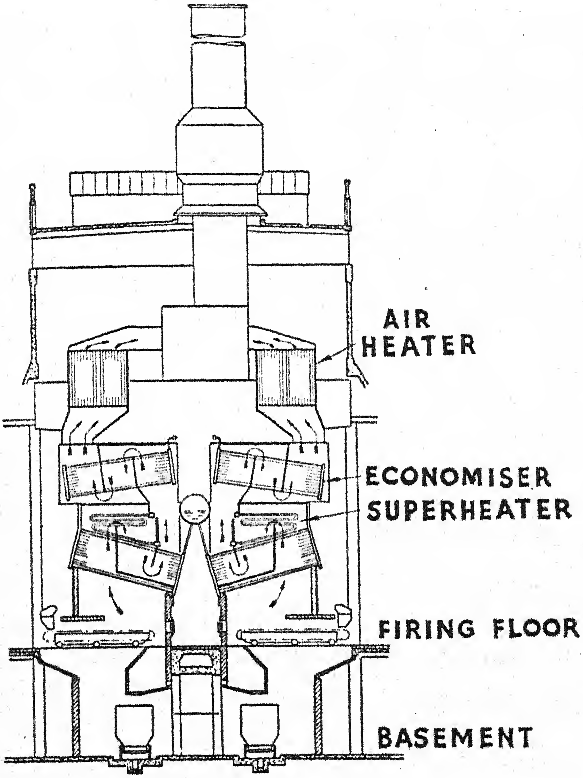


FIG. 154.

heated, that is, it contains more heat and has a higher temperature than saturated steam at the same pressure. If t_s is the temperature of the superheated steam, and t the temperature of saturated steam at the same pressure, the

extra heat added to the steam $= s(t_s - t)$, where s is the mean specific heat of the steam. The value of s varies according to the temperature, but an average value of 0·55 may be taken for approximate calculations with superheats of 100° F. to 200° F.

The considerable saving in both fuel and water, which may be obtained by using superheated steam instead of saturated steam, has led to its increasing use for steam engines and steam turbines.

Superheated steam may be produced either by adding heat to saturated steam after it leaves the boiler, which is the general method, or by reducing its pressure without doing work, as when steam passes a throttling valve. The latter method gives only a small amount of superheat, and there is also a loss of pressure.

The direct application of heat to saturated steam for the purpose of superheating it may be arranged in various ways, the method adopted in any particular case depending largely on the amount of superheat required and the type of boiler in which the superheating is to take place. For a high degree of superheat it is necessary to pass the steam through pipes surrounded by gases at a *high* temperature. Where the heat from the waste gases of a boiler are utilised, the degree of superheat will be *low*, as the gases themselves are at a comparatively low temperature. The increased efficiency of the plant obtained by using the heat in the waste gases is twofold : (1) there is a slight improvement in the boiler efficiency due to the utilisation of the waste heat, and (2) a small economy in the steam engine due to the use of steam which has been slightly superheated. Where steam is highly superheated by using the hot boiler gases, the boiler efficiency will probably not be affected, as the heat used for superheating the steam would otherwise have been used in producing saturated steam. The increased economy of the plant is due to the reduced weight of steam required by the engine.

By reducing the steam consumption of an engine of given horsepower we reduce the size of boiler required,

also the size of steam pipe required to convey the steam to the engine. Since, in the case of a condensing engine, a smaller weight of steam is delivered to the condenser, the size of the condensing plant is also smaller. Thus superheating not only effects a saving in fuel costs but also in capital cost of a steam plant.

Types of Superheaters.—All superheaters should be designed and constructed so as to give a rapid transfer of heat from the gas to the steam, be easily cleaned if necessary, and be free from the danger of being burnt out. Where the temperature of the gases does not exceed about 1000° F. there is no danger of burnt tubes, but for higher temperatures arrangements must be made to protect the tubes when no steam is passing through them, as, for instance, when steam is being raised, or when the engine supplied has a temporary stoppage.

Figs. 151 and 152 show the superheater and its position in a Babcock and Wilcox boiler. It is made of steel tubes 2 in. in diameter expanded into forged steel headers. The gases from the furnace pass across the water tubes once before meeting the tubes of the superheater, which is thus protected from the intense initial heat of the furnace. The tubes are prevented from being overheated when no steam is passing, by flooding them with water through a small pipe, shown at the right-hand side of Fig. 152. A small valve is used to drain away this water from the superheater tubes before steam is admitted to the engine. Any sudden stoppage of the engine is provided for by having a safety valve on the outlet pipe from the superheater, which is arranged to blow off before the main safety valve. In this way the surplus steam blows off through the superheater, and prevents the burning out of the tubes.

Schmidt Superheater.—Fig. 155 shows the Schmidt type of superheater as used for locomotives, and fitted at the smoke-box end of the boiler. Three of the upper rows of smoke tubes are made larger in diameter than the remainder of the tubes, and a superheater element is placed in each enlarged tube. Each element of the superheater consists of

a double loop tube, so arranged that the steam in passing through it traverses the large smoke tube four times. The arrows show the direction of the steam. The hot gases pass

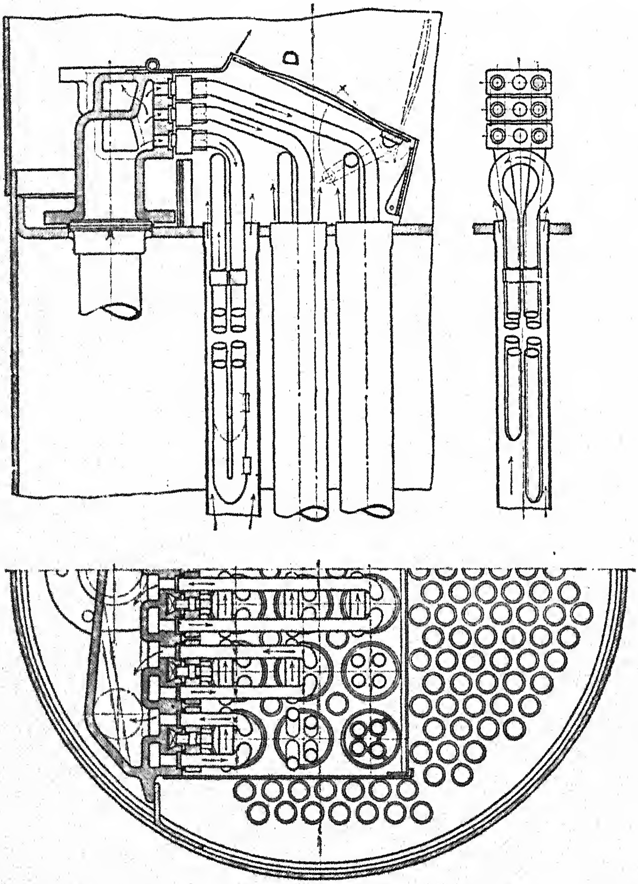


FIG. 155

through the large tubes and outside the double loop tube. The burning out of the superheater when the engine is not taking steam is prevented by a having a damper D (shown closed), which closes the smoke-box end of the large tubes,

thus preventing the passage of hot gases through them. The damper is opened and closed with the regulator valve; the damper can also be adjusted independently to give a less degree of superheat if required. This type of superheater gives a high degree of superheat.

Where highly superheated steam is used in an engine, drop valves or piston valves have been found to work very satisfactorily. Slide valves are less suitable, owing to the difficulty, at least in some cases, of lubricating the sliding surfaces.

The extent of the economy obtained by the use of superheated steam varies, but may be taken to give a saving of from 15 to 25 per cent. of the fuel in most cases. The author found in some experiments[1] on superheated steam that the initial condensation at cut-off was reduced 1 per cent. for every 7·5° of superheat. This number will, of course, vary somewhat with the type of engine using superheated steam. The reduction in steam consumption in the steam turbine is about 1 per cent. for every 10° of superheat. This is mainly due to the reduction of blade and disc friction owing to the density of superheated steam being much lower than the average density of saturated or wet steam.

Safety Valves

The safety valve provides for the safety of boilers by allowing the steam to escape when its pressure exceeds a certain limit. The safety valve is kept in its place on its seating either by a weight at the end of a lever, by a strong spring, or by a heavy weight, placed directly over the valve, and these three forms will here be described.

A good safety valve is one which will not permit the pressure in the boiler to rise above a fixed point, and, having reached that point, will allow all excess of steam to escape as fast as it is generated by the boiler.

Mr. Webb, of the London and North-Western Railway,

[1] *Proc. Inst. C.E.*, vol. cxxviii.

in an experiment on a locomotive boiler fired hard, found that a pipe $1\frac{1}{4}$ in. diameter was sufficient to allow all the steam to escape as fast as generated without the pressure increasing beyond the initial pressure.

The Lever Safety Valve.—This valve (Fig. 156) rests on a circular brass seating, and is prevented from rising by the steam pressure underneath the valve, by the weight at the end of the lever. The disadvantage of this valve is that it

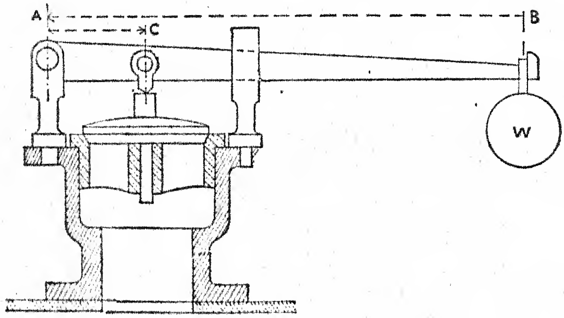


FIG. 156

admits of being tampered with, and the effect of a small addition to the weight is magnified considerably in its action on the valve.

To find the weight W, or length of lever AB, for a given pressure of steam :

Let AB=L=length of lever from fulcrum A to centre of weight W.

AC=c=distance between centre of valve and fulcrum.

W=weight at end of lever.

w=weight of lever acting at centre of gravity of lever, assumed at one-third from large end.

P=pressure of steam per square inch.

a=area of valve.

V=weight of valve.

(1) If the effect of the weights of valve and lever be omitted, we have, when valve is just about to lift—

Moment of downward pressures

=moment of upward pressures

$$W \times L = Pa \times c$$

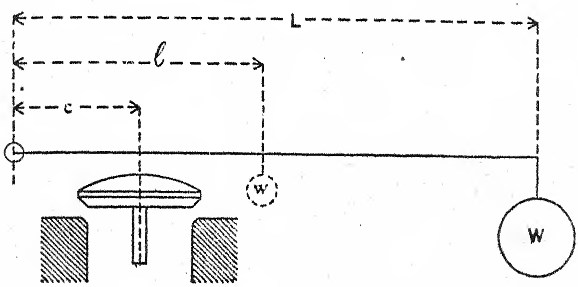


FIG. 157.

(2) Taking the effects of weights of lever and valve into account, we have, when valve is about to lift—

$$WL + wl = (Pa - V)c$$

Example. Let it be required to find weight W at end of lever when $L=36$ in. ; $c=4\frac{1}{2}$ in. ; $l=\frac{1}{3}L$; $w=10$ lb. ; $V=2$ lb. ; $P=100$ lb. ; $a=5$ sq. in.

(1) Omitting weight of valve and lever—

$$\begin{aligned} W \times L &= Pa \times c \\ 36W &= (100 \times 5)4{\cdot}5 \\ &= 62{\cdot}5 \text{ lb.} \end{aligned}$$

(2) Including the weight of valve and lever—

$$\begin{aligned} WL + \frac{wL}{3} &= (Pa - V)c \\ 36W + \frac{10 \times 36}{3} &= (500 - 2)4{\cdot}5 \\ &= 58{\cdot}9 \text{ lb.} \end{aligned}$$

These results show that if a weight of 62·5 lb. were placed on the lever instead of 58·9 lb., the valve would not lift at 100 lb. pressure as required, but at a somewhat higher pressure.

Fig. 158 shows an improved form of lever safety valve by Messrs. Yates and Thom. In this design knife-edges are substituted for pins, thus reducing friction and increasing sensitiveness, and the lever is so shaped that a horizontal line passes through all the points of application of the knife-edges, thus placing the lever in a condition of stable equilibrium.

The Dead-weight Safety Valve.—Fig. 159 illustrates a

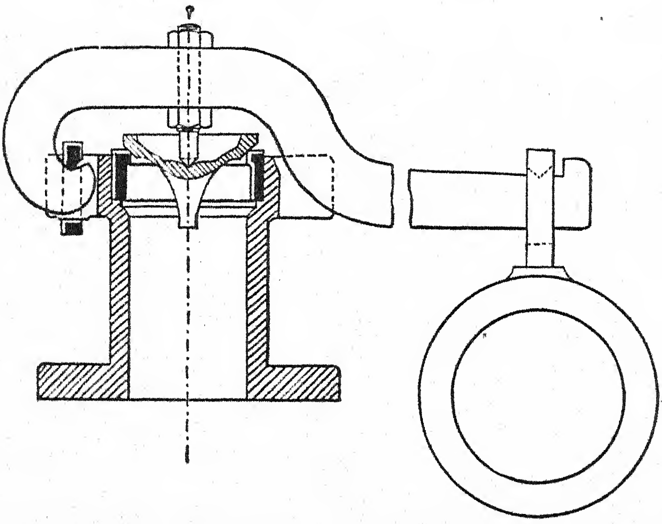


FIG. 158.

dead-weight safety valve as used for stationary boilers. The valve *a* rests on the seating *b*, which is fixed on the top of a long pipe, as shown. The valve is secured to a large casting A, which fits down over the pipe like a cap. This casting is provided with a ledge on which circular rings of metal, which act as weights, may rest.

To find the dead weight required (including casting and weights) for a valve of given area : Multiply the area of the valve by the pressure per square inch at which the valve

is required to lift. Thus a valve 3 in. diameter to blow off at 100 lb. pressure requires the following dead weights :

$$\text{area} \times \text{pressure per square inch} = 3 \times 3 \times 0{\cdot}7854 \times 100$$
$$= 706{\cdot}86 \text{ lb. dead weight.}$$

Spring-loaded Safety Valve.—For locomotives and marine engines both the lever and dead-weight types are unsuitable for obvious reasons, and the valve must be spring loaded,

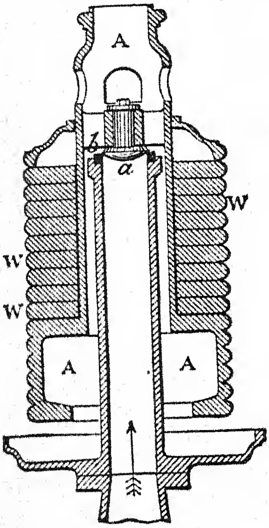


FIG. 159.

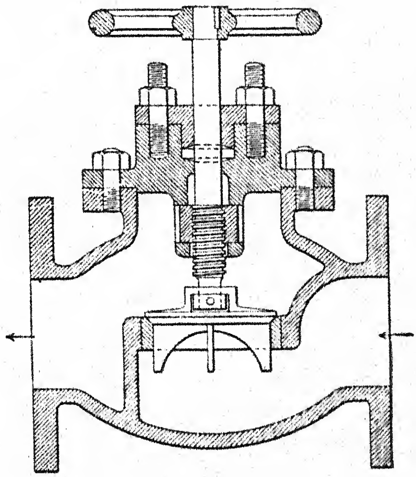


FIG. 160.

as such a valve is unaffected by vibration or deviations from the vertical.

The following diagram, Fig. 161, illustrates what is known as Ramsbottom's safety valve. It consists of two separate valves and seatings having one lever, bearing on the two valves, and loaded by a spring, the spring being placed between the valves. The tension on the spring can be adjusted by the nuts. By pulling or raising the lever the driver can relieve the pressure from either valve separately. and ascertain that it is not sticking on the seating.

One disadvantage of the spring-loaded safety valve is that the load on the valve increases as the valve lifts, so that the pressure required just to lift the valve is less than

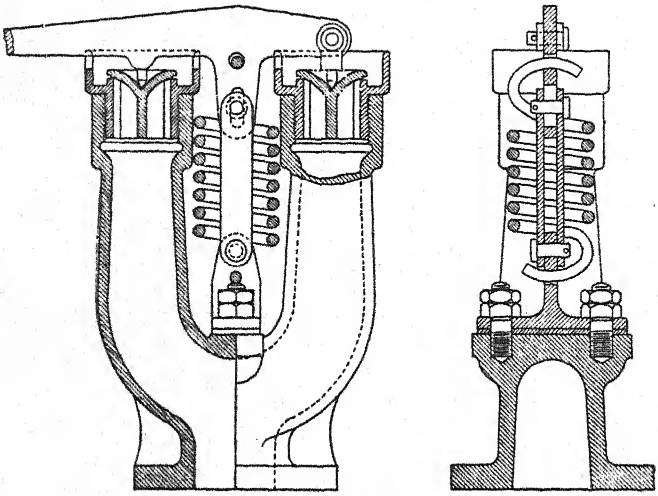


FIG. 161.

that required to open it fully. For this reason in some cases it is arranged that the area acted on by the steam is greater when the valve is open than when the valve is closed. Fig. 162 shows one way of accomplishing this. When the valve is closed the steam acts on a diameter *d*, but when the valve opens the pressure acts on the larger diameter D. Such valves are very rapid acting.

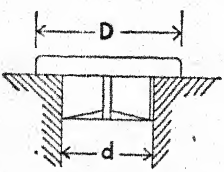


FIG. 162.

Steam Regulating Valves.—The stop valve is used to open or close the communication between the boiler and engine. A common form of valve is shown in Fig. 160. It consists of a valve which may be opened or closed by means of a screwed spindle which is turned by a hand wheel.

The *gridiron valve* (Fig. 163) is an arrangement for giving a large opening to the passage of steam with a comparatively small travel of the valve. It consists of a flat valve composed of a number of bars which move on a seating, having

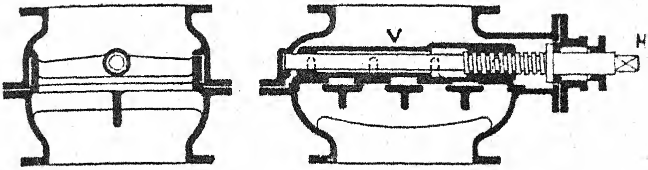


FIG. 163.

a number of ports or openings which are covered by the valve, as shown in the figure.

The valve V is opened or closed by the screw turned by a handle at H.

The equilibrium double-beat valve (Fig. 164) consists of two disc valves, A and B, on one spindle, each of which has its own seating. The arrows show the direction of the steam on entering the valve box from the passage E. The valve B is made a little larger than A, to enable the valve A to be put in its place from the top. The pressure acts on the top of one valve and on the bottom of the other, hence the two valves are nearly in equilibrium and may be easily lifted from their seating when under pressure.

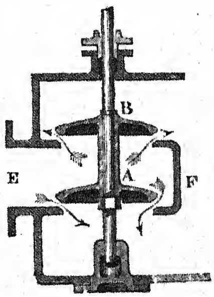


FIG. 164.

This arrangement provides a large opening to steam with valves of comparatively small diameter.

BOURDON'S PRESSURE GAUGE

The pressure gauge registers the pressure of steam in the boiler above the pressure of the atmosphere. The following

figure (Fig. 165) illustrates the construction of Bourdon's gauge, which is the one commonly used.

The gauge consists of a curved tube, BB, of a flattened or elliptical cross-section, as shown enlarged at C. The tube is closed at one end and open at the other, by which the interior of the tube is put in communication with the boiler pressure through the cock A. The closed end of the tube is attached (as shown in the figure) to a sector D, provided with teeth which gear with those of a small pinion

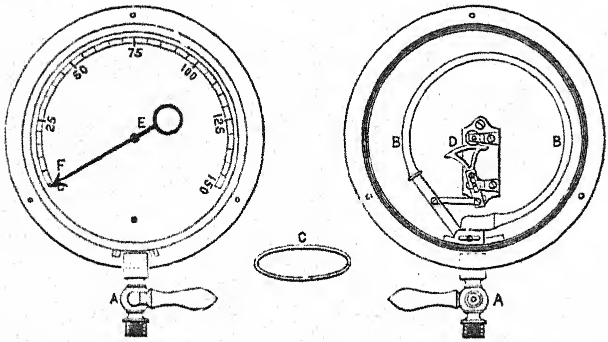


FIG. 165.

on the same axis as the finger EF on the face. The effect of steam pressure in the curved tube is that the tube *tends to straighten itself*, and thus, as the pressure increases, the closed end moves the sector, which acts on the finger and indicates the pressure. These gauges are carefully graduated by comparing their indications with those of a mercurial gauge.

If the zero only of the gauge is in error, this can be corrected by shifting the pointer on its pin. If the gauge is reading too high or too low after the zero has been corrected, the rate of magnification can be corrected by altering the position of the fixed pin in the slot shown in the sector actuating the pinion.

Auxiliary Plant

The Separator.—The steam produced by a boiler may be either wet, dry, or superheated; but in any case there will be a loss of heat from it during its passage through the steam pipe from the boiler to the engine tending to produce wetness. The use of wet steam in an engine or turbine is uneconomical besides involving some risk; hence it is usual to endeavour to separate any water that may be present from the steam before the latter enters the engine. Many devices for effecting this separation are in use; a common principle of construction being to suddenly change the direction of the flowing mixture of steam and water. The result of this is that the water, being denser than the steam, continues in the original direction, and thus separation is effected. Fig. 166 shows one form of separator. The current of steam is reversed in direction, and the water in the steam is deposited in the large chamber C. A gauge glass is fixed at G to show the height of water in the chamber. A steam trap may be fixed to remove the water through the pipe P. The velocity of the steam is reduced by making the area of the separator larger than the steam pipe; this prevents the steam from removing the water already in the separator.

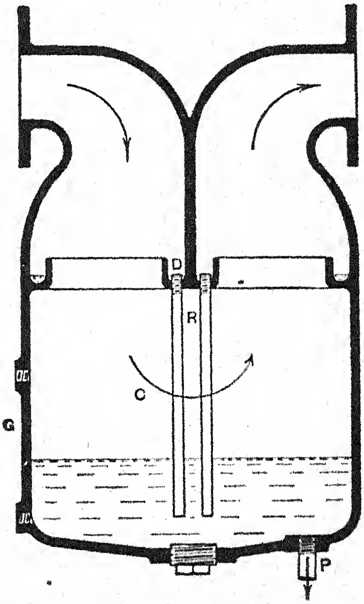


Fig. 166.

Any water deposited on the sides of the pipe and carried forward by the steam current, is caught at D and drained by the pipes R. The steam may pass through the separator in either direction.

Separators should be efficient, that is, they should dry the steam with little or no reduction of the pressure of the steam passing through them, and should be of sufficient capacity to deal with a sudden influx of water.

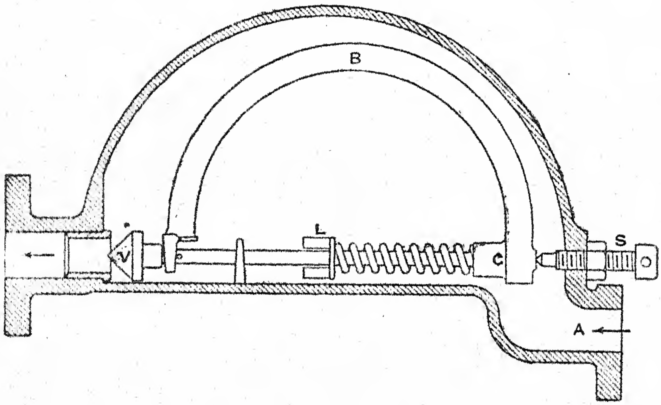


FIG. 167.

Some separators rapidly lose their efficiency at high steam velocities.

Steam Trap.—The water separated from the steam by the separator may be automatically drained away by means of a steam trap.

Fig. 167 shows in section an expansion steam trap (Sirius type). A hollow spring tube B of nickel steel contains a liquid which becomes a gas at temperatures above the lowest that the steam is likely to have. The water enters at A, and, being lower in temperature than the steam, it causes the tube B to contract and the valve V is opened. When all the water is discharged and steam enters the trap, the increased temperature converts the liquid in

the tube B into gas, the curved tube tends to straighten itself, and so to close the valve V.

The trap can be arranged to discharge either continuously or intermittently by the adjusting screw S. The end C of the tube B is held against the adjusting screw by the spring, one end of which presses against the lugs L fixed to the casing.

Feed Heaters.—Before steam can be obtained from water it is necessary to raise the water to the temperature at which evaporation will take place. By heating the cold feed water before it enters the boiler there is less heating to be done in the boiler itself, and also a saving of fuel if the feed water is heated in the first instance by waste heat. The admission of hot feed water instead of cold feed water to the boiler tends to prevent any strain on the boiler, such as obtains when there is local cooling of a portion of the hot shell of the boiler. A further important advantage which follows the use of the feed water heater is the separation of most of the dissolved gases, such as air and carbon dioxide present in the water, which is liberated by heating ; also any calcium bicarbonate present in solution is decomposed on heating, a portion of its carbon dioxide being driven off, and the remaining calcium carbonate remains as an insoluble substance, and may be removed before entering the boiler. The tubes in a feed heater (Fig. 168) are liable to become covered with grease from the exhaust steam and with scale from the feed water. This would reduce their efficiency for transmitting heat, and arrangements should be made for periodical cleaning of the heater.

The heat for heating the feed water may be obtained either from the chimney gases, or from the exhaust steam, or by using live steam from the boiler itself.

The use of the chimney gases for heating the feed water is explained on p. 172, where the construction of an economiser is described. Fig. 168 shows the construction of an exhaust steam feed heater. The cold feed water enters at A and fills the internal body of the feed heater, the hot feed water passes out at B. The exhaust steam from the engine

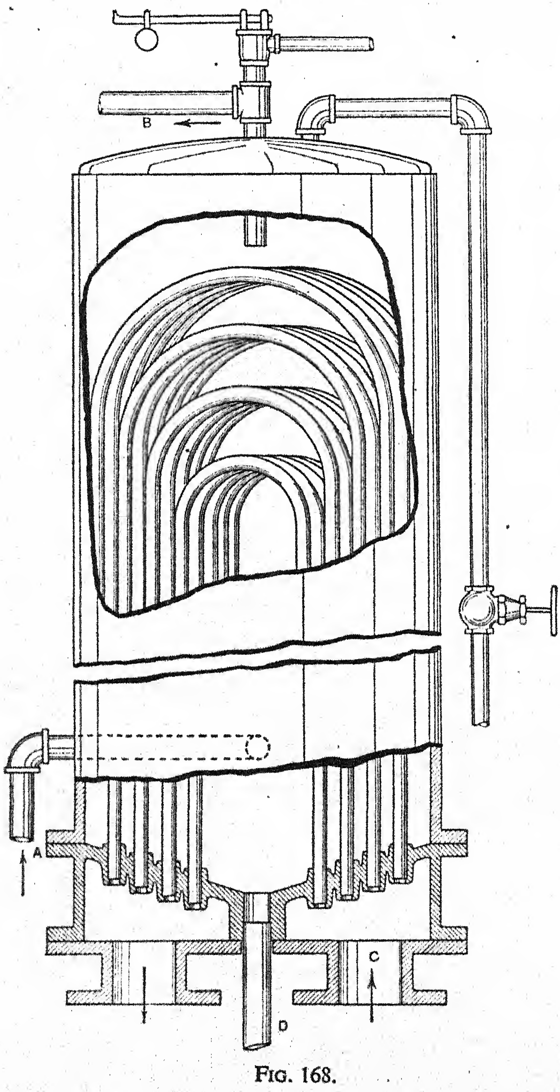


FIG. 168.

enters at C and passes through the curved tubes, leaving the vessel at E. Any solid matter in the feed water may be withdrawn at D.

The use of boiler steam to heat the feed water would not appear at first to offer any economic advantage, but experience shows that apart from the general advantages obtained by feed heating, there is a small gain in economy with some types of tank boiler, mainly due to improved circulation.

CHAPTER XVI

FUELS AND COMBUSTION

The principal fuels used by engineers are coal and oil, and the chief constituents of these are carbon and hydrogen.

Atmospheric air consists of two invisible gases, oxygen and nitrogen, in the proportion of 23 parts of oxygen and 77 parts of nitrogen in every 100 parts of air by weight. These gases are not united in any way, they are merely mixed together. The oxygen is the active element in air, and it is ready to unite with anything for which it has affinity, providing the surrounding temperature is raised sufficiently high to enable it to do so. All fuels contain elements which readily unite with oxygen. The nitrogen of the air takes no part whatever in the process of combustion, and merely serves to dilute the oxygen. The process of combustion may be easily understood by considering the case of an ordinary gas flame.

When we wish to 'light the gas'—that is, to set in operation the process of combustion, or chemical union between the oxygen of the air and the carbon and hydrogen of the gas—we have first of all to apply heat with a match ; otherwise, if the tap is turned on, the gas will escape, but it will not burn. Once started, however, the burning proceeds vigorously and uniformly, and results in the evolution of heat. Before the escaping gas was lighted we could detect the strongly characteristic odour of unburnt coal gas, but no such odour can be detected from the burning gas flame. The reason of this is that the carbon and hydrogen of the coal gas have united with the oxygen of the air to form two odourless and invisible compounds, namely, carbon dioxide (CO_2) and steam (H_2O).

The combustion of coal differs, however, from the case just considered ; for, when coal is thrown on a furnace,

there are three distinct stages in its combustion : first, the gases contained in the coal are distilled off as in the ordinary process of gas making ; secondly, these gases are either consumed or pass up the chimney unconsumed ; thirdly, the remaining solid residue of the coal is burnt. Considering the gases distilled from the coal, which consist principally of marsh gas (CH_4) and olefiant gas (C_2H_4) : in order that they may be completely burnt, (1) they must be thoroughly mixed with a sufficient supply of oxygen ; (2) the temperature of the mixed gases must be sufficiently high to allow of chemical combination taking place.

When the distilled gases from the coal are not mixed with a sufficient supply of oxygen, or the temperature is not sufficiently high, then clouds of finely divided carbon are disengaged from the gas, and pass up the chimney in the form of smoke, part of which is deposited in the flues as soot. But if the disengaged carbon is supplied with sufficient oxygen, and the temperature is sufficiently high for ignition or combination to take place, it burns with a bright flame.

Considering the solid fuel which remains as coke or carbon, it should be explained that carbon is capable of forming two different compounds with oxygen, namely, carbon monoxide (chemical symbol, CO) and carbon dioxide (chemical symbol, CO_2), depending on the abundance of the supply of oxygen to the carbon during the process of combustion.

When the supply of oxygen is sufficient, and is intimately mixed with the fuel in the presence of a sufficiently high temperature, the carbon is completely burnt to CO_2 ; but when there is an insufficient supply of oxygen, or the oxygen is not intimately mixed with the fuel, then CO is formed. The effect of this on the production of heat may be seen by the table on page 210.

When the air for combustion in a boiler furnace passes between the fire bars under the fuel, combination takes place between the oxygen and the under layers of glowing carbon, forming CO_2. This gas, in passing on through the

II. Table of Heat of Combustion

Combustible	Heat of combustion per lb. (B.Th.U.)	Lb. of water evaporated from and at 212°.
Hydrogen	62,000	64·2
Carbon burned to carbonic oxide .	4,400	4·55
Carbon burned to carbonic acid .	14,600	15·0
Anthracite	14,700	15·2
Newport coal	14,000	14·5
Durham coke	13,640	14·1
Wigan cannel coal	14,000	14·5
Petroleum	20,360	21·0
Oak wood (dried)	7,700	8

upper layers of carbon, here loses part of its oxygen, and the CO_2 is now reduced to CO, the remainder of its oxygen having united with more carbon to form CO.

If now sufficient air is supplied at the surface of the fuel, this carbonic oxide will burn with a blue flame, with further evolution of heat ; but if it is not so supplied, it will pass up the chimney unconsumed, and the difference between the heat of complete and incomplete combustion of carbon, namely, 10,200 B.Th.U. per pound of carbon, will be lost.

Fuel.—The more commonly used fuels consist of coal, coke, and oil.

Coals are of various qualities and compositions, but they may be roughly divided into two classes : anthracite and bituminous coals.

Anthracite consists almost entirely of pure carbon, and it contains little volatile matter. It therefore burns with very little flame and without smoke. It gives a very intense local heat in the furnace when fully ignited, but it ignites with difficulty and burns slowly. It is necessary to burn it as a thin fire on a large grate area and with a strong draught. It is much less effective as a steam-raising coal than bituminous coal.

Bituminous coals contain, besides carbon, a large proportion of more or less volatile matter as hydrocarbons, which

are distilled off from the coal by the action of heat in the furnace.

The quality and composition of these coals vary considerably, but when of good quality they burn freely, with a long flame.

Coal containing volatile hydrocarbons is good steam-raising coal, but some qualities of it emit much black smoke unless the furnace is very carefully handled, and it also sometimes cakes and becomes pasty.

Coke is made by driving off the volatile matter in bituminous coal. As the coal is heated in the absence of air, no combustion takes place. The volatile matter, containing *coal gas* and tarry substances, is purified and stored in gasometers for lighting, heating, and power. The remaining *coke* is a mixture mainly of carbon and ash.

Coke burns without smoke, and (like anthracite) produces a strong local heat in the furnace, transmitting a large proportion of its heat by radiation through that portion of the plate immediately over the furnace. The grate area should be large and the fire thin.

Ash and Clinker. The relative value and suitability of coals for steam raising depend in some measure upon the proportion of ash they contain, and on the nature of the ash, whether or not it tends to form clinker upon the bars.

Ash is the non-combustible material remaining behind after the combustible material in the coal has been burned. Some kinds of ash tend to fuse and to form a semi-liquid silicate of iron and alumina, called *clinker*, which flows in a molten state over the bars, choking and obstructing the air passages between the bars, and deadening the fire unless it is removed. The more rapid the combustion the more rapidly clinker accumulates.

Calorific Value of a Fuel.—This represents the total number of heat units evolved by the complete combustion of 1 lb. of fuel.

The numerical value of the total heat for any given fuel is determined by experiment, by burning a weighed sample

of the fuel in a calorimeter, immersed in or surrounded by water.

The fuel is burned by being supplied with oxygen under pressure, or by being mixed with a chemical compound rich in oxygen.

The total heat of the combustion is absorbed by the water, and its amount in heat units is measured by noting the rise in temperature of the known weight of water surrounding the calorimeter.

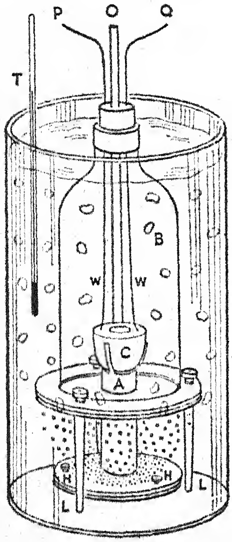


FIG. 169.

Fig. 169 illustrates Darling's calorimeter, as used for the calorific value of solid fuels. The combustion takes place inside a glass bell B (a copper bell can be substituted if greater accuracy is required). The fuel is placed in the crucible C and ignited by means of a thin wire attached to the ends of the brass leads W, connected to the battery at P and Q. A stream of oxygen under slight pressure is delivered to the fuel through the tube O, the pressure of the oxygen being sufficient to prevent water entering the bell B. The products of combustion pass downward at A and bubble upward through the water by which B is surrounded, giving up their heat to the water.

Use of Calorimeter.—1. Carefully grind up an average sample in an iron mortar, and weigh out 1 to 1·5 grams in the crucible C. Brush any particles from the sides into the mass at the bottom.

2. Prepare a quantity of water at a temperature about 2·5° C. below the temperature of the room. Water drawn from a tap varies in temperature according to the season ; but usually it will be necessary to add a little warm water

to it in order to bring it to the requisite temperature. Measure out 1400 c.c. into the vessel.

3. Place the crucible in position, and fasten the glass cover down upon the rubber ring by means of the screws, so as to form an airtight joint. The screws must only be turned until a resistance is felt ; any further tightening might crack the glass cover.

4. Insert the rubber cork into the neck so that the ignition wire is embedded in the fuel. The copper wire W should terminate about level with the rim of the crucible, and the tube O delivering the oxygen about $\frac{1}{2}$ in. above the surface of the fuel.

5. Turn on a gentle stream of oxygen from a cylinder or gas holder, and immerse the apparatus in the water. Carefully note the temperature of the water, and complete the battery circuit. As soon as the fuel is ignited, disconnect the battery. Allow the combustion to proceed steadily until completed, then continue passing the oxygen, mixing the water by lifting the combustion arrangement up and down, until no further rise of temperature is observed. Note the temperature carefully.

PRECAUTIONS : (*a*) The oxygen must never be admitted so rapidly as to cause particles to be blown out of the crucible. The time required to burn 1 gram of average coal in a steady stream of oxygen is about 5 minutes.

(*b*) If the sample is observed to burn with a smoky flame, the combustion must be stopped, as the result will be valueless owing to unburnt carbon. A second combustion should then be performed with the end of the oxygen tube 1 in. below the cork until all volatile matter has burnt off, after which the tube may be pushed down to the crucible and the combustion completed.

(*c*) During the combustion the tube delivering the oxygen should be moved about so as to ensure that every particle of coal is consumed. The flexibility of the rubber cork allows of this operation being easily

performed. In every case near the end of the combustion, the supply of oxygen should be sufficient to cause the crucible to become visibly red hot, whereby complete combustion is secured.

6. The calorific value is calculated as follows :

$$\frac{(\text{Weight of water}+{}^{1}\text{Water equivalent})\times\text{Rise of Temp.}}{\text{Weight of fuel taken}}=\text{Calorific value}$$

1 c.c. of water is taken as weighing 1 gram. If a Fahrenheit thermometer is used, the result will express British thermal units per pound of fuel ; if Centigrade readings are taken, the figure will represent either calories per gram or pound—degree C units per pound.

Example.—1 gram of Welsh steam coal burnt as above.
Water taken = 1400 c.c. or grams.
Water equivalent of apparatus and vessel } = 204 grams.
Temperature of water before combustion = 14° C.
Temperature of water after combustion = 19·20° C.
Temperature of room = 16·6° C.

$$\text{Calorific value}=\frac{(1400+204)\times(19{\cdot}20-14)}{1}=8341 \text{ cals. per gram.}$$

or 8341 lb.—° C. units per pound.
$8341\times\frac{9}{5}=15{,}014$ B.Th.U. per pound.

The calorific value may be calculated from the following formula when the chemical composition of the fuel is known :

$$\text{Calorific value (B.Th.U. per lb.)}=14{,}600\,C+62{,}000\left(H-\frac{O}{8}\right)$$

where C, H, and O stand for the weights of carbon, hydrogen, and oxygen respectively contained in 1 lb. of fuel. The other elements are neglected.

The latter part of the formula is based on the assumption that the oxygen present in the fuel is not free oxygen, but

[1] The figure expressing the 'water equivalent' of the calorimeter and vessel is furnished with the apparatus. It is the weight of water which would absorb the same amount of heat as the apparatus.

is already united with the hydrogen as water, and that, therefore, the total hydrogen available for combustion is reduced by an amount equal to one-eighth the amount of oxygen present (because one-eighth the weight of oxygen in water represents the weight of hydrogen present).

Example.—A sample of Newcastle coal has the following composition: $C=0{\cdot}78$, $H=0{\cdot}052$, $O=0{\cdot}086$. Then its total heat of combustion

$$=(14{,}600\times 0{\cdot}78)+62{,}000\left(0{\cdot}052-\frac{0{\cdot}086}{8}\right)=13{,}900 \text{ B.Th.U.}$$

The above formula assumes that the calorific value of a sample of fuel is the sum of the calorific values of the elements composing the fuel, but actual determinations by a calorimeter give values varying from 4 to 10 per cent. lower than the value calculated in this way. This is mainly due to the fact that some heat is absorbed in splitting up the complex hydrocarbons and gasifying the fuel. Other formulæ have been proposed (by Goutal and others) which give closer results in some cases but not in others, and students are warned that if accurate results are required in calculating thermal efficiencies the *measured* value must always be taken.

Higher and Lower Calorific Values.—When the calorific value of a fuel is measured by means of a fuel calorimeter, the products of combustion are cooled down to approximately atmospheric temperature, so that any steam formed due to combustion of hydrogen is condensed and adds its latent heat to the measured value. In actual practice the flue gases of a boiler or the exhaust gases of an internal combustion engine pass away at so high a temperature that the steam is not condensed and so its latent heat is not available. The *useful* calorific value is thus lower than the value measured as above by the latent heat of the steam formed as a result of combustion. This value is known as the Lower Calorific Value (L.C.V.), while the value obtained without this deduction is known as the Higher Calorific Value (H.C.V.).

In the above example the calculated value of 13,900

B.Th.U. is the H.C.V. The weight of steam formed is $9 \times 0{\cdot}052 = 0{\cdot}468$ lb., and since the latent heat is 967 B.Th.U. per lb. the amount to be deducted is 450 B.Th.U. Thus the L.C.V. is $13{,}900 - 450 = 13{,}550$ B.Th.U. (3·2 per cent. less). In the case of fuel oil the difference may be as much as 12 per cent.

In looking through the results of engine and boiler trials care should be taken to ascertain whether the thermal efficiency is based on the H.C.V. or the L.C.V.

Example.—In a test, a fuel oil was found to have a gross or higher calorific value of 18,500 B.Th.U. per lb. : water vapour was formed by combustion at the rate of 1·08 lb. of vapour per pound of oil burnt. What is the net or lower calorific value of the oil ?

Solution.—It is usual to cool the products of combustion, during a test of calorific value, down to about 100° F. Hence each pound of water vapour yields up 967 units of latent heat on condensation, and 112 units of sensible heat on further cooling, which would not be available under ordinary conditions of work. The total $= 967 + 112 = 1079$, say 1080 B.Th.U. per pound of water formed.

In the example given the weight of vapour formed per pound of oil $= 1{\cdot}08$ lb., and the heat given up by it $= 1{\cdot}08 \times 1080 = 1166$ B.Th.U.

The lower calorific value of the fuel is therefore

$$18{,}500 - 1166 = \underline{17{,}334 \text{ B.Th.U. per lb.}}$$

Loss of Heat in the Chimney Gases.—Suppose 1 lb. of coal whose calorific value is 14,600 B.Th.U. is burned in a boiler furnace by the aid of 20 lb. of air, then (neglecting ash) 21 lb. of gaseous products pass away through the flues and up the chimney, carrying away with them the following amount of heat to waste : Let the temperature of the air entering the furnace be 60° F., and the temperature of the gases leaving the boiler-heating surface be 600° F. ; then, taking 0·24 as the average specific heat of the gaseous products—

$$21 \times (600 - 60) \times 0{\cdot}24 = 2721{\cdot}6 \text{ B.Th.U.,}$$

$$\text{or} \quad 2721{\cdot}6 \div 14{,}600 \times 100 = 18{\cdot}8 \text{ per cent. waste heat}$$

Weight of Air required for Combustion.—The minimum weight of air required per 1 lb. of fuel burned will depend upon the chemical composition of the fuel. Thus, if the fuel

were pure carbon, and the carbon were burned to CO_2, then—

	C	+	O_2	=	CO_2
	Carbon		Oxygen		Carbon dioxide
Molecular weight	12	+	2×16	=	44
or	1	+	$2\frac{2}{3}$	=	$3\frac{2}{3}$

that is, 1 lb. of carbon burned to CO_2 requires $2\frac{2}{3}$ lb. of oxygen to complete the combustion ; or, since in every hundred parts of air by weight, twenty-three parts are oxygen, 1 lb. of carbon burned to CO_2 requires $(2\frac{2}{3} \times \frac{100}{23})$ $= 11{\cdot}6$ lb. of air.

Completing the above equation in terms of *air* supplied in pounds per pound of carbon—

$$\text{Carbon} + \text{air} = CO_2 + \text{nitrogen.}$$
$$1 \quad + \quad 11{\cdot}6 = 3{\cdot}6 \quad + \quad 9$$

The hydrogen present in the coal combines with the oxygen to form steam (H_2O). The following chemical equation represents the combustion of hydrogen to H_2O.

$$H_2 \quad + \quad O \quad = H_2O$$
$$\text{hydrogen} + \text{oxygen} = \text{steam}$$

or, 2 lb. of hydrogen combine with 16 lb. of oxygen to produce 18 lb. of steam. Therefore 1 lb. of hydrogen requires 8 lb. of oxygen for its complete combustion. Weight of air required $= 8 \times \frac{100}{23} = 34{\cdot}7$ lb.

Example.—Find the minimum amount of air required to burn completely 1 lb. of fuel consisting of 0·82 lb. of carbon and 0·05 lb. of hydrogen.

$$\text{Air required} = 0{\cdot}82 \times 11{\cdot}6 + 0{\cdot}05 \times 34{\cdot}7 = 11{\cdot}18 \text{ lb.}$$

In practice an excess of air up to 50 per cent. is necessary to ensure complete combustion, owing to the difficulty of ensuring satisfactory mixing of the air with the fuel. Without this excess, the loss of heat from incomplete combustion would be much greater than the loss due to excess air.

However, it is most important that no more air is supplied than is absolutely necessary to ensure complete combustion, since each pound of air entering at, say, 60° F. and leaving at 600° F. carries away $0{\cdot}24(600-60) = 130$ B.Th.U.

This is about 1 per cent. of the calorific value of an average coal, so if we supply 10 lb. of air more than really necessary per pound of coal the boiler efficiency is reduced by 10 per cent., a very important consideration when large quantities of coal are being burned.

Suppose the case where double the theoretical quantity of air is supplied, then the equation becomes—

$$\text{Carbon} + \text{air} = CO_2 + \text{nitrogen} + \text{free oxygen.}$$

$$\underbrace{1 + 23{\cdot}2}_{24{\cdot}2} \qquad \underbrace{3{\cdot}6 + 18 + 2{\cdot}6}_{24{\cdot}2}$$

(Compare the previous equations.)

We can here find the percentage of CO_2 and oxygen by weight passing away in the chimney gases. Thus—

(i) $3{\cdot}6 \div 24{\cdot}2 \times 100 = 14{\cdot}9$ per cent. of CO_2.

(ii) $2{\cdot}6 \div 24{\cdot}2 \times 100 = 10{\cdot}8$ per cent. of oxygen.

It will be noticed from the above equation that if n lb. of air be supplied to the furnace per 1 lb. of carbon, there are always $(n+1)$ lb. of gases of various composition passing away at the chimney.

Also assuming that all the carbon is burned to CO_2, then for every 1 lb. of carbon burned there are 3·6 lb. of CO_2, neither more nor less, whatever the weight of air supplied ; but the greater the excess of air the smaller the *percentage* of CO_2 becomes, and the greater, also, the percentage of free oxygen.

Again, if the percentage composition of the chimney gases be determined by chemical analysis, it is possible to find from it by calculation the weight of air supplied to the furnace per pound of coal. A chemist might make the chemical analysis, but the engineer should be able to make this calculation for himself.

To calculate the Weight of Air supplied to the Furnace per Pound of Coal.

Example.—Given that the average composition of a boiler flue gases by volume is 9·5 per cent. CO_2, 1·5 per cent. CO, and 7 per cent. oxygen, find the weight of air supplied per pound of coal (containing 80 per cent. carbon).

NOTE.—Chemical analysis usually finds the percentage composition of the gases by *volume.*

(1) The first step is to convert relative volumes into relative weights, and for this purpose we find the relative densities of the various gases.

In this connection it is necessary to note that the density of elementary gases is proportional to their atomic weight, while the density of compound gases is proportional to half their molecular weight ; thus the molecular weight of $CO_2=44$, for—

$$CO+O_2=CO_2$$
$$12+32=44$$

Therefore the relative density of $CO_2=\frac{\text{molecular weight}}{2}$

$$=\frac{44}{2}=22$$

Similarly for CO—

$$C+O=CO$$
$$12+16=28$$

Relative density of $CO=\frac{\text{molecular weight}}{2}$

$$=\frac{28}{2}=14$$

(2) *To find the total parts by weight of oxygen and carbon present.*—Referring to the actual percentages given in the problem, we have—

Gas	Percentage of gas by volume		Relative density		Parts by weight
CO_2 . . .	9·5	×	22	=	209
CO . . .	1·5	×	14	=	21
O . . .	7·0	×	16	=	112
					342

Of all the air supplied to the furnace per 1 lb. of carbon, we have here an account of the whole of the oxygen, part of it being united to the carbon to form CO_2, part to form CO, and the remainder being present as free oxygen.

Also the whole of the carbon originally in the coal is now present, combined with oxygen either as CO_2 or CO.

(3) *To find the weight of oxygen present per pound of carbon.*—Having obtained the proportional parts by weight of the constituent gases, we can now find the weight of oxygen present per pound of carbon, and from this the weight of air supplied ; thus—

$$C+O_2=CO_2$$
$$12+32=55$$

Simplifying—

$$3+8=11$$
$$\text{or } \tfrac{3}{11}+\tfrac{8}{11}=1$$

showing the weight of carbon and oxygen per pound of CO_2.

Also
$$C+O=CO$$
$$12+16=28$$
$$3+4=7$$
$$\tfrac{3}{7}+\tfrac{4}{7}=1$$

showing the weight of carbon and oxygen present per pound of CO ; then—

		C.	O.
209 parts of CO_2 contain	$209\times\frac{3}{11}$ of C =	57	—
,, ,,	$209\times\frac{8}{11}$ of O =		152
21 parts of CO contain	$21\times\frac{3}{7}$ of C =	9	—
,, ,,	$21\times\frac{4}{7}$ of O =		12
112 parts of oxygen	=		112
	Totals . .	66	276

That is, 276 parts by weight of oxygen have been supplied for 66 parts by weight of carbon : or pounds of oxygen per pound of carbon

$$=276\div 66=4{\cdot}18 \text{ lb.}$$

(4) *To find the weight of air per pound of carbon.*—

$$4{\cdot}18\times\tfrac{100}{23}=18{\cdot}2 \text{ lb. of air.}$$

(5) *To find the weight of air per pound of coal.*—Having found the air per pound of carbon, since the coal contains by the problem 80 per cent. of carbon, then weight of air per pound of coal

$$=18{\cdot}2\times\tfrac{80}{100}=14{\cdot}56 \text{ lb. per pound of coal.}$$

Loss due to the presence of CO in the chimney gases.—The presence of even a small percentage of CO in chimney gases is a sure indication of imperfect combustion, due generally to insufficiency of air or imperfect mixing of the gases. The result is that the carbon in burning to CO only produces 4400 heat units instead of 14,600 heat units, which would have been produced if the carbon had been completely burned to CO_2.

To calculate the percentage loss due to the presence of CO, the same example given above may be used. From the calculated table there given, we see that 57 parts of C were burned to CO_2, and 9 parts of C were burned to CO.

Now, the number of heat units produced by the burning of 57 lb. of carbon to CO_2

$$=57 \times 14{,}600 = 832{,}200 \text{ B.Th.U.}$$

and the heat units produced by the burning of 9 lb. of carbon to CO

$$=9 \times 4400 = 39{,}600 \text{ B.Th.U.}$$

Or a total of $832{,}200 + 39{,}600 = 871{,}800$ B.Th.U.

But if the whole of the $(57+9)$ lb. of carbon had been burned to CO_2, we should have had—

$$66 \times 14{,}600 = 963{,}600 \text{ B.Th.U.}$$

Hence the loss due to the presence of only 1·5 per cent. of CO—

$$=\frac{963{,}600-871{,}800}{963{,}600} \times 100 = 9{\cdot}52 \text{ per cent.}$$

Another method of finding approximately the weight of air supplied per pound of coal may be adopted when the flue gases are passed through an economiser (Figs. 140 to 143).

Let w=weight of feed water per hour passing through the economiser, and g the weight of flue gases per hour passing through the flues ; then, taking 0·24 as the specific heat of the flue gases, and measuring the respective values of t_1 and t_2 for the feed water, and t_3 and t_4 for the gases by thermometers, we have—

$$\underset{\textit{Heat gained by water}}{w(t_2-t_1)} = \underset{\textit{Heat lost by gases}}{g(t_3-t_4) \times 0{\cdot}24}$$

from which g, or the weight of gases passing through the flues per hour, may be obtained, w being known.

From g, the weight of gases, subtract k, the weight of coal burned per hour ; then the remainder $(g-k)$ repre-

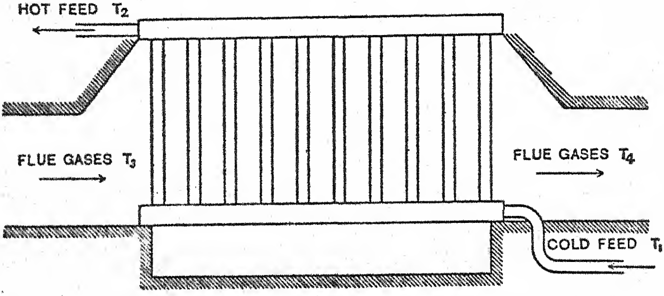


FIG. 170.

sents the weight of air supplied per hour, and $(g-k) \div k$ =pounds of air supplied per pound of coal.

Example.—In a Lancashire boiler, to which an economiser is attached, the weight of coal burned per hour is 600 lb. ; the weight of feed water passing through the economiser is 5000 lb. per hour ; the temperature t_1 of the feed water entering the economiser is 100° F., and the temperature t_2 of the feed water leaving the economiser is 220° F. ; the temperature t_3 of the flue gases entering the economiser is 600° F., and the temperature t_4 of the gases leaving the economiser is 300° F. Find the weight of air supplied to the boiler per pound of coal.

$$w(t_2-t_1)=g(t_3-t_4)\times 0{\cdot}24$$
$$5000(220-100)=g(600-300)\times 0{\cdot}24$$
$$g=8333 \text{ lb.}$$

Then, since as above—

$$(g-k)\div k=\text{weight of air required,}$$
$$(8333-600)\div 600=12{\cdot}9 \text{ lb. of air per pound of coal.}$$

This method is only approximate, because it takes no account of the heat absorbed by and radiated from the brickwork of the economiser.

CHAPTER XVII

THE FURNACE

Temperature of Combustion.—There is a distinction to be drawn between the *quantity* of heat resulting from combustion and the *intensity* of the heat, for, having a given quantity of heat measured by temperature difference passing through the flues, it is upon the intensity of the heat that the efficiency of the furnace depends, the transmission of heat from the furnace to the water being proportional to the difference of temperature on the two sides of the boiler-heating surface.

The same *quantity* of heat will be evolved by the complete combustion of 1 lb. of fuel whether the fuel is burned with a minimum of air or in a large excess of air, also whether it is burned in one minute or one hour. But the *intensity* or temperature of combustion in the two cases will be very different, and therefore the rate of steam-production will be very different. The method of calculating the theoretical maximum temperature is as follows : Suppose 1 lb. of carbon to be completely burned in 12 lb. of air, which is about the minimum theoretical quantity : then the products of combustion are 13 lb. Taking 0·24 lb. as the specific heat of the products, and allowing 14,600 heat units per pound of carbon, we have—

$$\frac{14{,}600}{13 \times 0{\cdot}24} = 4647^\circ \text{ F.}$$

This temperature as above calculated is never realised in actual practice for various reasons.

In the first place, there is usually at least 50 per cent. excess air, which would lower the calculated temperature to $\frac{2}{3} \times 4647 = 3100^\circ$ F., and in the second place, since the

combustion takes time, heat is being radiated and conducted to the comparatively cold surfaces while combustion is taking place. Brickwork in and about the furnace, by storing up heat, is an aid to combustion and to high temperature. It is important, however, that the brickwork should not hide effective heating surface.

Natural draught is draught induced by the aid of a chimney only, and without the assistance of any mechanical appliance such as a steam jet or fan. The draught is caused by the difference of temperature, and therefore of density, between the hot column of gas in the chimney and a similar column of cold air outside the chimney.

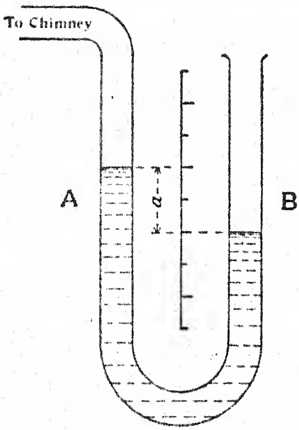


FIG. 171.

A chimney is a costly structure, and when considered as a means of producing draught, is by no means the most efficient way of doing it. But a tall chimney of some kind is necessary, in any case, to carry off the gaseous products of combustion, and to prevent, as far as possible, solid particles from escaping into the air. It is also convenient, in ordinary cases, to utilise it at the same time as a means of producing a draught.

When a keener draught is required, then some system of mechanical draught is adopted. The draught is usually measured by taking the difference in level between the surfaces of two columns of water in the two legs of a U-tube, one leg being connected with the chimney, and the other open to the air. Thus the difference *a* (Fig. 171) between the water levels in A and B is a measure of the draught in inches of water head. This difference represents a very small amount of actual effective pressure ; thus in ordinary chimneys the pull of the chimney is represented

by about $\frac{1}{2}$ in. of water in small chimneys to $\frac{3}{4}$ in. in high chimneys.

$$
\begin{array}{llll}
\text{Since } 2{\cdot}3 \text{ ft. head of water} & & =1 \text{ lb. pressure} & \\
\quad 27{\cdot}6 \text{ in.} & ,, & =1 \text{ lb.} & ,, \\
\quad 1 \quad ,, & ,, & =\frac{1}{27{\cdot}6} \text{ lb.} & ,, \\
\quad \frac{1}{2} \quad ,, & ,, & =\frac{1}{55{\cdot}2} \text{ lb.} & ,,
\end{array}
$$

The amount of draught which can be produced by a chimney having a height of h feet depends upon the absolute mean temperature T_1 of the hot gas in the chimney and the absolute temperature T_2 of the cold air outside the chimney.

The amount of draught which can be produced by a chimney can be calculated when the temperature of the gas in the chimney and that of the outside air are known.

Consider a chimney to be 100 ft. high and to have an area of 1 sq. ft. Let the gas in the chimney have an average temperature of 600° F., and let the temperature of the outside air be 60° F. The weight of 1 cu. ft. of gas or air may be taken as 0·0807 lb. per cu. ft. at 32° F. The weight of 1 cu. ft. of gas or air varies inversely as its absolute temperature.

Therefore the weight of 100 cu. ft. of air at 60° F.

$$=\frac{32+461}{60+461}\times 100\times 0{\cdot}0807=7{\cdot}64 \text{ lb.}$$

Similarly the weight of 100 cu. ft. of gas at 600° F.

$$=\frac{32+461}{600+461}\times 100\times 0{\cdot}0807=3{\cdot}75 \text{ lb.}$$

The difference in pressure between these columns of cold air and hot gas$=7{\cdot}64-3{\cdot}75=3{\cdot}89$ lb. per sq. ft.

This difference of pressure may be converted to inches of water, which is the usual method of expressing boiler draught. A cubic foot of water produces a pressure of 62·3 lb. per sq. ft. and its height is 12 in.

8

Therefore the height of water required to produce a pressure of 3·89 lb. per sq. ft.

$$=\frac{3\cdot 89}{62\cdot 3}\times 12=0\cdot 75 \text{ in. of water.}$$

To make the best use of the Chimney Draught.—The most common cause of serious loss of efficiency of the boiler is the flow of cold air to the chimney other than through or over the fire. This may happen in various ways.

(1) By the too common practice of admitting large volumes of air at the back of the bridge through the ashpit instead of through the fire (see Fig. 172). This system is

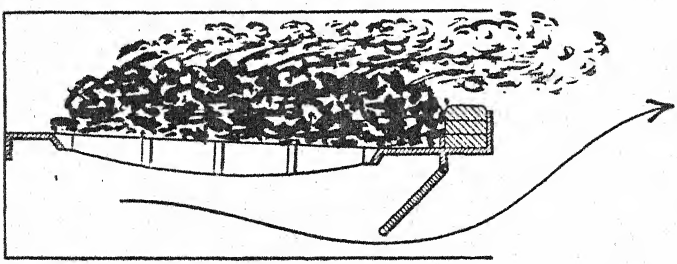


FIG. 172.—Conditions bad: furnace overcharged with coal; cold air in large volume escaping to flues by the ashpit; little air passing *through* the fire; temperature of furnace low.

sometimes adopted to prevent smoke, but where adopted it should only be used sparingly, and be open for flow of air during that portion of the time only when black smoke is being produced—that is, for a few seconds or minutes immediately after firing.

(2) By leaky, cracked, and badly fitting brickwork. A lighted taper passed over the surface of the brickwork all round the outside of the boiler and flues will often reveal serious and unexpected leakages of air, which are more or less spoiling the draught through the fire, cooling the flue gases, and thereby reducing the efficiency of the heating surface, and carrying away heat to the chimney which would otherwise have been used to generate steam.

(3) By the easy flow of air to the flues through imper-

fectly covered fire bars, the fires having been allowed to burn into holes, and especially to burn hollow near the bridge. This portion of the grate requires frequent attention.

The stronger the draught the thicker the fire should be, but with a given chimney draught, and assuming a constant quality of the fuel, there is a certain thickness of fuel on the grate bars which will give the best results, that is, will give a maximum temperature of the furnace, and it is the chief duty of the fireman to secure and maintain as uniformly as possible this condition of maximum temperature.

If a furnace is fed at first with an excess of coal, making

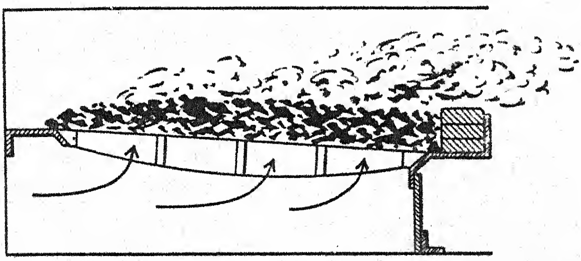


FIG. 173.—Conditions good: air passing *through* fire to flues; maximum temperature of furnace; fuel incandescent; fresh fuel added in light charges; supplementary air supply through adjustable grid in fire-door for few minutes to burn smoke.

the fire too thick, then less air will pass through the fuel than is necessary for perfect combustion, owing to the increased obstruction by the fuel of the flow of air ; the furnace temperature will be low, and unburned or imperfectly burned fuel in the form of carbon monoxide and hydrocarbon gases—evidenced by the presence of clouds of smoke—will pass away to the chimney.

But as this fire burns down—if its thickness is kept even, and no hollow places are allowed to form in it—the furnace temperature gradually increases, until at a certain thickness of the fire a state of brilliant white incandescence of the fuel is reached, which represents the condition of maximum temperature, and therefore of maximum effectiveness of the

fire. This is the condition at which a balance is found between the air supply and the coal burned to give the greatest possible temperature of the furnace, and therefore also the maximum rate of steam generation.

The following data are given by M. Pouillet as illustrating the way in which the temperature of a fire may be judged by its appearance :

Appearance	Temp. Fahr.	Appearance	Temp. Fahr.
Dull red . .	1290°	Orange deep .	2010°
Cherry dull . .	1470°	,, clear .	2190°
,, full . .	1650°	White heat . .	2370°
,, clear .	1830°	,, dazzling .	2730°

When the maximum temperature has been attained, the fire should then be kept as nearly as possible uniformly in this condition by frequent thinly spread charges of fuel. The volatile gases given off from the newly fired charge will probably require, for a short period, a supplementary air supply through an adjustable opening in the fire door or other equivalent.

It should, however, be noted that all admission of air to the flues other than through the fire reduces the tendency of the air to flow through the fire against the resistance of the bed of fuel, and therefore reduces the rate of combustion of the solid fuel upon the grate bars.

When a fire is uneven in thickness the air flows unevenly through it, the flow being more swift through the thin parts. The result is a rapid tendency of the fire to burn into holes at the thin places (Fig. 174). The fire at the bridge end of the grate should be kept a little thicker than elsewhere, because of the tendency of the draught to find the shortest way to the chimney, and particularly to burn the fire hollow close to the bridge.

In a range of furnaces all connected to one common chimney, if one or more of these fires burns thin or into

hollow places, the air rushes through these grates as an easy path to the chimney, the effect of which is that not only are the flues cooled, but the draught through the remaining thicker fires is much reduced, the steam pressure will fall, and bad and wasteful conditions will prevail.

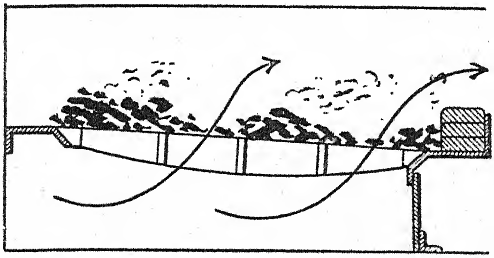


FIG. 174.—Conditions bad: cold air entering flues through hollow places without taking part in combustion of coal.

When in any case the draught is stronger than is needed to burn the weight of coal required, the draught may be regulated by partially closing the damper or by reducing

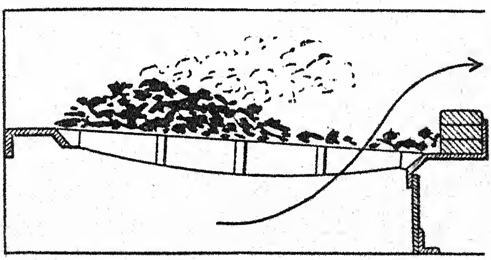


FIG. 175.—Conditions bad: fire looks well from front, but burnt out and hollow at back; cold air escaping to flue.

the area of the grate by brickwork, and then working the fires thicker on the bars.

If *smoke* is to be prevented, the causes of its formation should be understood. Smoke consists of particles of unconsumed carbon and, apart from the atmospheric contamination produced, its presence leads to a loss of economy

for two reasons. In the first place, such carbon as is unconsumed represents a loss of heat. In the second place, an accumulation of particles of carbon in the form of soot on the heating surfaces reduces their capacity for absorbing heat and the gases pass away from the boiler at a higher temperature, thus taking away heat that might be used for evaporation with clean surfaces. Unfortunately, the presence of smoke only represents a few per cent. of the heating value of the coal, and where coal is comparatively cheap and the installation is not a large one, it often is cheaper to allow smoke than to pay the cost of preventing its formation. It must be understood that it is difficult and in most cases impossible to consume smoke once it is formed. 'Smoke-consuming' appliances are usually 'smoke-preventing' appliances.

When a fresh charge of coal is thrown on to a glowing fire the first thing that happens is that volatile matter (coal gas and tarry vapours) is distilled off. If (*a*) there is insufficient air present to burn this or (*b*) if the air is not well mixed with the volatile matter or (*c*) if the temperature is not sufficiently high, the combustion will be incomplete, some of the unconsumed carbon settling out in the form of smoke or soot. These facts enable us to trace some of the causes of smoke and suggest some means of reducing or avoiding smoke.

(1) If a large quantity of coal is fed to the furnace quickly, large quantities of volatile matter are liberated, and the ordinary air supply is insufficient to cope with this. The result is incomplete combustion and smoke.

(2) Unless the oxygen supplied with the air has *access* to the carbon and hydrogen of the fuel, combustion cannot take place. An ordinary bunsen flame is a good example of this. If air is admitted through the holes at the bottom of the bunsen tube it mixes very thoroughly with the gas and a faint blue flame, indicating complete combustion, is the result. If the holes are closed air has access only to the external surface of the gas issuing from the tube, so that combustion only takes place at this surface, the heat

of combustion raising the unconsumed hydrocarbons in the interior to a temperature which produces a luminous flame. If such a flame is brought into contact with a cold surface carbon will be deposited on the surface. For this reason a luminous flame in a boiler furnace should not be brought into contact with relatively cold plates or tubes before combustion is complete, or smoke is almost certain to result. A draught of cold air through the fire door may produce the same result.

(3) The mere mixing of air with gas is not sufficient to produce combustion. The mixture must be at a high temperature. If, for instance, the fire is covered with fresh coal, insufficient heat is radiated to the mixture of air and gas above the fire, so that incomplete combustion and smoke will result.

It is a good plan to throw the coal at one time over the right-hand half of the grate, and the next time of firing to throw the coal over the left-hand half, so that there is always one half of the fire more or less in a state of incandescence, and smoke from the newly fired half is to a large extent obviated ; this is called *alternate-side firing*.

Under some conditions *coking stoking* is preferred, that is, the fresh coal is thrown on the front part of the grate, and then from time to time it is pushed bodily towards the back. In this way the smoke from the front of the fire is consumed as it passes over the incandescent fire at the back of the grate.

That fireman is most efficient who maintains the highest average temperature of the furnace, and this is by no means always obtained by the man who burns the most coal.

Methods of Avoiding Smoke.

1. Small charges of coal fed frequently to the furnace. With hand firing this would involve opening the furnace door too frequently, but with mechanical stokers fuel can be fed either continuously or at frequent short intervals.

2. As much of the air as possible to be fed *through* the fire. This has the double effect of heating the air and splitting it up into small streams, ensuring thorough mixing

with the gases given off by the coal. With thick fires it is necessary to admit some air *above* the fire, owing to the 'producer' effect of a thick fire.

3. Keep the mixture of gas and air out of contact with cold surfaces until combustion is complete. A firebrick arch, heated by the furnace, over which the gases pass, helps to maintain a high temperature and prevent smoke.

Rate of Combustion of Coal in Boiler Furnaces.—This rate is measured by the number of pounds of coal burned per hour per square foot of grate surface. It varies in proportion to the draught available, and to the quality of the fuel, from 10 to 24 lb. per sq. ft. of grate per hour in Lancashire boilers with natural draught, to as high as from 80 to 160 lb. of coal per square foot of grate per hour with forced draught in locomotive and water-tube boilers.

Accelerated Draught.—In order to increase the rate of combustion in steam boilers, and thus to increase the evaporative capacity or power of the boiler, various systems are adopted for accelerating the draught to the furnace.

The earliest example of this is the exhaust steam blast still used in the chimney of the locomotive, and the effect of which is that the rate of coal consumption is three or four times as great as would be the case without the blast. By this means a comparatively small and light boiler is capable of great evaporative capacity—qualities which are indispensable for railway purposes, where great tractive power is required with the least possible wear and tear on the bridges and permanent way.

Accelerated draught may be either *forced* or *induced.* Forced draught is obtained by means of a steam blast or fans delivering air to the furnace under a pressure measured in inches of water varying from 1 in. to 3 in. or more.

Fig. 176 illustrates a Meldrum apparatus, which provides a simple means of obtaining a forced draught, and which is much used for burning an inferior quality of very small coal, which requires a keen draught to burn it. The apparatus consists of two blowers fixed on the front plate of a closed ashpit, and projecting under the fire bars.

The blowers are tubes which expand trumpet-shape towards the inner end. A small steam jet at the front end of the tube induces a strong flow of air at a high velocity into the ashpit, thereby causing an air pressure under the grate bars. These blowers, may be regulated for any desired rate of combustion, from 16 or 18 lb. to 28 lb. or more per square foot of grate.

The use of steam jets for this purpose is undesirable, as

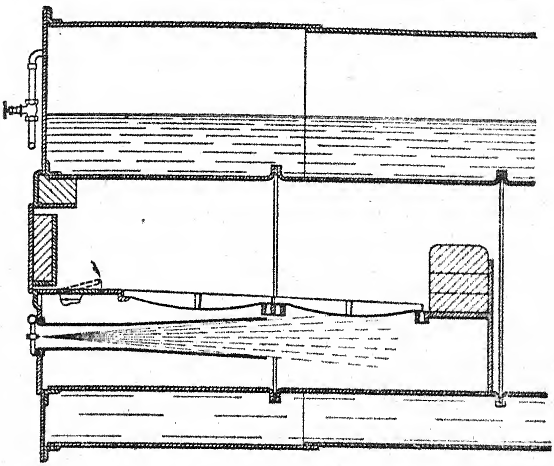


FIG. 176.

the tendency is toward excessive steam consumption, owing to wear of the jet orifices.

Induced draught is draught obtained by drawing air through the fire and flues by means of a fan at the base of the chimney. The fan is made large enough to receive the whole of the gaseous products of the flues, and it delivers them to the chimney. The vacuum formed in the flues by the suction action of the fan sets up a difference of pressure between the flue and the outside air, and causes the air to flow through the fire into the flues at any desired rate depending on the speed of the fan.

With induced draught the fan is working with hot gases

and tends to depreciate more rapidly, but there is the advantage that it is not necessary to close in the front of the boiler furnace as with forced draught. The pressure in the boiler is below atmospheric so that air tends to leak in through defective brickwork, etc. With forced draught any leakage is outward.

In some cases a system of *balanced draught* is used, in which a forced draught fan maintains a pressure in the ashpit and an induced draught fan maintains a slight vacuum in the flues. These are so balanced that the pressure above the furnace is atmospheric, so that if the firing door is opened air neither flows in nor out.

In all cases of accelerated draught, whether forced or induced, it is necessary to increase the thickness of the bed of fuel on the grate, so as to correctly proportion the weight of fuel burnt to the weight of air supplied, otherwise there may be a large loss of heat at the chimney by excess of air passing through the fire.

Here, as before, the object is to secure and maintain a maximum temperature of the furnace. Especial care must be taken with accelerated draught to avoid holes and hollow places in the fire.

Mechanical Stokers.—The coal may be placed on the fire by mechanical means instead of by hand firing. Most of the various types of mechanical stokers attempt to imitate hand firing, by either spreading the coal uniformly over the fire in small quantities at a time or by placing the fresh coal at the front of the grate so as to produce the coking system of firing. The fuel is supplied to the grate at a uniform rate from a coal hopper, and with the aid of self-cleaning bars, no opening of the fire door is necessary. The latter operation admits large volumes of cold air and reduces the efficiency of hand-fired grates. Mechanical stokers usually have some form of movable grate, which maintains the fires clear.

The labour required for stoking the fires is reduced, and a higher efficiency is claimed to follow the use of mechanical stokers. It is also found possible to burn a cheaper coal

than can be burned with hand firing. Where the steam required is steady and continuous, mechanical stokers are working under the best conditions, but where the demand for steam is of a fluctuating character, mechanical stoking is not always sufficiently flexible.

A type of stoker which does not attempt to imitate hand firing is the under-feed stoker. Instead of placing the fresh coal on the top of the fire as is the usual plan, the fresh coal is forced underneath the incandescent coal already on the grate. The hydrocarbon gases given off pass through the hot coal, and are there burnt. This type of stoker requires the use of a coal containing little ash and clinker-forming material.

The frontispiece shows a chain grate mechanical stoker applied to a Stirling boiler. The grate is in the form of an endless link chain, and moves slowly forward. The coal falls from the hopper on to the front of the grate, where it is coked, and is then carried forward at such a rate as to completely burn the coal before the grate returns. The ashes fall into the ashpit under the grate, and may be removed by a spiral ash conveyor or by hand. The rate at which the grate moves can be mechanically adjusted to suit the character of the coal or the rate of firing. The framework supporting the grate rests on four wheels, which arrangement permits of ready withdrawal of the grate for inspection or repairs.

A modern development is the use of coal in a pulverised or powdered form. In this case no grate is required, the fuel being blown into the combustion chamber mixed with air. In its powdered form the fuel presents an enormous surface for a given volume, so that combustion is extremely rapid and a smaller excess of air can be used. The system is more suitable for large water-tube boilers but can be, and has been, applied to tank-type boilers. The cost and upkeep of the pulverising plant is rather high for small installations.

Liquid Fuel.—The great development of the oil resources of the world, and the many advantages possessed by oil as

a fuel, has led to its increasing use for the generation of steam in steam boilers as well as being directly employed to generate power in the cylinders of oil engines.

Oil possesses many advantages over coal as a fuel, though at the present time it possesses one great disadvantage, namely, that it is usually much more expensive than coal. The following are some of the advantages of oil :

1. It possesses a greater heat value per pound than coal, giving about 19,000 B.Th.U. per pound as against a maximum of 14,500 B.Th.U. per pound for coal, or over 30 per cent. higher. Hence a given weight of oil will enable a vessel to travel at least a proportionately greater distance than with an equal weight of coal without requiring a fresh supply of fuel.

2. There is less waste with oil, there being no clinker or ash to be removed.

3. Oil fuel may be quickly stored on board ship through a pipe with a minimum amount of labour and an absence of dust. It can be stored in the double bottom, leaving the usual coal bunker space for other purposes.

4. Less firemen are required in the stoke hole.

5. There is no cleaning of fires, no opening of fire doors, causing admission of large volumes of cold air. There is, therefore, a more constant temperature in the furnace and less wear and tear of the boilers.

6. Since very thorough mixing of the fuel oil with air can be obtained with modern types of burners, a smaller air excess can be used, giving higher flame temperature and higher efficiency than with coal firing.

On the other hand, the cost of fuel oil per ton is much higher than that of coal, so that the actual fuel cost per pound of steam generated is appreciably greater when oil fuel is used.

Oil, as obtained from the different oil fields, varies greatly in its composition. Crude oils from some oil fields contain considerable quantities of the lighter oils, lubricating oil, paraffin and petrol, which may be driven off from the residual oil by the process of distillation. The

heavy residual oils are those most suitable for raising steam in boiler furnaces. The absence of the lighter oils in the residual oil makes its storage perfectly safe.

Combustion of Oil.—Various methods are employed in order to obtain a complete combustion of the oil in the furnace. The conditions are that the oil shall be atomised, that is, broken up into minute particles and then raised to a high temperature in the presence of a sufficient supply of air. The oil may be atomised either directly or by the aid

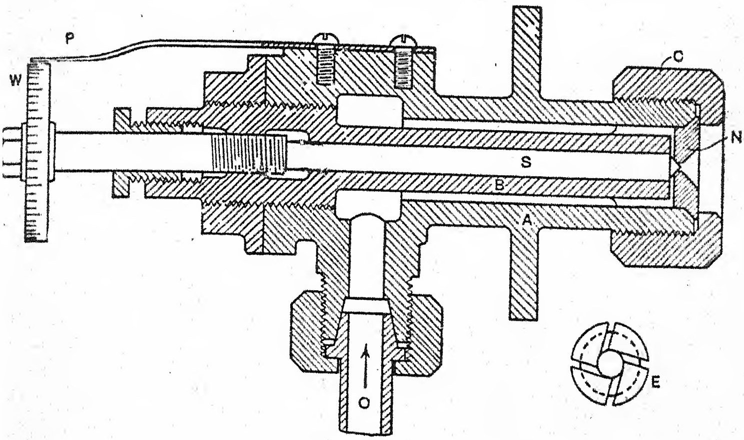


FIG. 177.

of compressed air, or by the use of steam, which should preferably be superheated to avoid the presence of water in the atomiser, which may have the effect of extinguishing the flame.

By compressing the oil to from 30 to 100 lb. per sq. in., and heating it to from 150° F. to 200° F. before entering the atomiser, the use of both air and steam may be dispensed with, and this appears to be the simplest method of burning the oil.

Care should be taken that the oil contains no dirt, by passing it through an efficient filter, otherwise the atomiser may become choked.

Fig. 177 illustrates the pressure type of burner as made

by Messrs. Kermode, Ltd. Oil enters the burner at O under pressure, and passes between the inner wall of the outer cylinder A and the outer wall of the plug B. The latter is in contact with the nozzle N. A number of grooves are cut along the plug B parallel to the axis, and also in the end of the plug as shown by the end view E. The end grooves are tangential to the end of the plug and give to the oil a rotary motion. The nozzle N is made of hard steel, and is held in position by the cap C. The oil issues from the burner as a fine spray in the form of a cone with its apex at the nozzle. The amount of oil passing to the burner is regulated by the hand wheel W, which moves the spindle S, and the pointer P indicates the degree of opening of the oil supply at the nozzle.

CHAPTER XVIII

THE STEAM TURBINE

If a jet of fluid (water, gas, or steam) issues from a nozzle at high speed and strikes a stationary plate it exerts a pressure on the plate due to the destruction of momentum, the force exerted being equal to the momentum destroyed per second. If the plate is moving away from the nozzle the force exerted is less but work is done since the plate is moving. If a series of such plates are mounted on the rim of a wheel we have a simple form of ' impulse ' turbine. The efficiency of such a turbine, i.e. the fraction of the energy supplied which is converted to work, would be low, and curved vanes would be necessary to enable the fluid to enter the vanes without shock and to utilise more energy by directing the fluid backward.

When fluid issues from a nozzle there is a backward reactive force on the nozzle equal to the amount of momentum generated per second in the nozzle. If a series of nozzles are mounted on the rim of a wheel and it is possible to arrange for fluid to enter the nozzles at a low velocity and leave at a high velocity, the nozzles will move backward and the wheel will do work. In this case we have a simple form of ' reaction ' turbine.

In the case of steam turbines none of these are purely ' impulse ' or purely ' reaction.' In each case steam is expanded from a given pressure to a lower pressure in nozzles or their equivalent, thus converting heat into kinetic energy. As much as possible of this kinetic energy (or velocity energy) is then converted into work in the moving blades.

It will be seen that while in the reciprocating steam engine the pressure of the steam is used directly, in the steam turbine the pressure energy is first converted into velocity energy and then used.

The class of turbine in which the steam is allowed to expand before entering the moving blades but not in them is called an *impulse* turbine. The De Laval and Rateau types are examples of impulse turbines.

Turbines, such as the Parsons turbine, in which the steam expands in moving blades as well as stationary blades (equivalents of nozzles) are known as *reaction* turbines. Actually there is a combination of impulse and reaction in this type.

In the *impulse* type there is a drop of pressure in the nozzles, but no drop of pressure through the moving blades. In the *reaction* type there is a drop of pressure in both stationary blades (acting as nozzles) and in moving blades. Each type has advantages and disadvantages as compared with the other and these will be contrasted later.

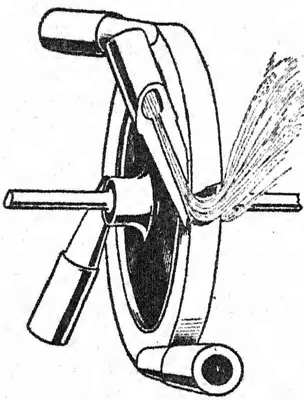


FIG. 178.

The De Laval turbine is a simple impulse turbine, in which the steam expands from boiler pressure to condenser pressure before entering the wheel containing the blades. The velocity thus attained by the steam is very high, being about 4000 ft. per second when the absolute pressure is 160 lb. per sq. in. on entering and the condenser pressure is 1 lb. per sq. in.

A general view of the De Laval wheel with four nozzles is shown in Fig. 178, in which a section of one of the expanding nozzles is also shown.

Fig. 179 shows the construction of a De Laval impulse wheel. The wheel is made solid and the shaft is bolted to the wheel. A wheel with a hole through the centre is much weaker than a solid wheel.

Fig. 180 shows the section of a De Laval nozzle and a shutting-off valve.

Fig. 181 shows the section of a De Laval turbine as made

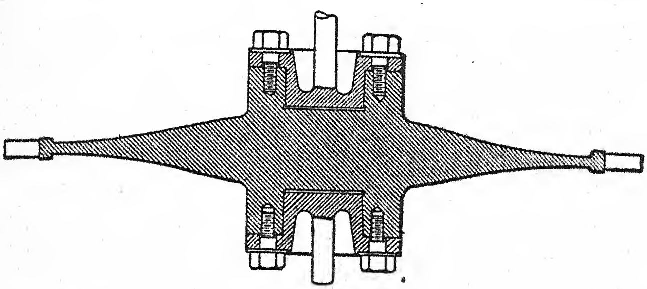


Fig. 179.

by Messrs. Greenwood and Batley, Leeds. The steam from the boiler passes through the stop valve C and then through the strainer D, to remove any solid material, which

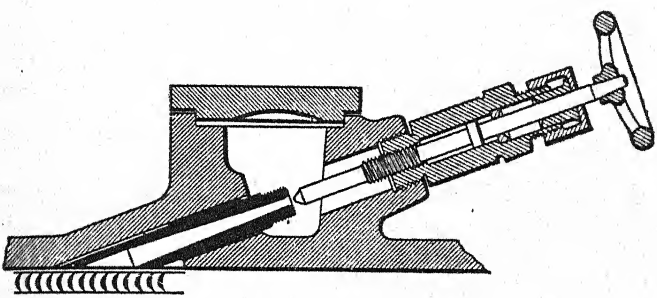


Fig. 180.

might injure the blades. The amount of steam admitted to the steam chest F is regulated by the governor valve E. From F the steam passes to the steam nozzles (see Fig. 182), and then meets the vanes in the turbine wheel G. After leaving the wheel, the steam is in the exhaust chamber H.

and is led to the atmosphere or to the condenser by the pipe J.

The turbine shaft is supported by the bushes *n* and *m*, and by the ball bush *w*, which rests on a spherical seat. To prevent the admission of air into the turbine when running condensing, a tightening bush is introduced at X. Most of the lubrication is by wick lubricators in the oil tank P.

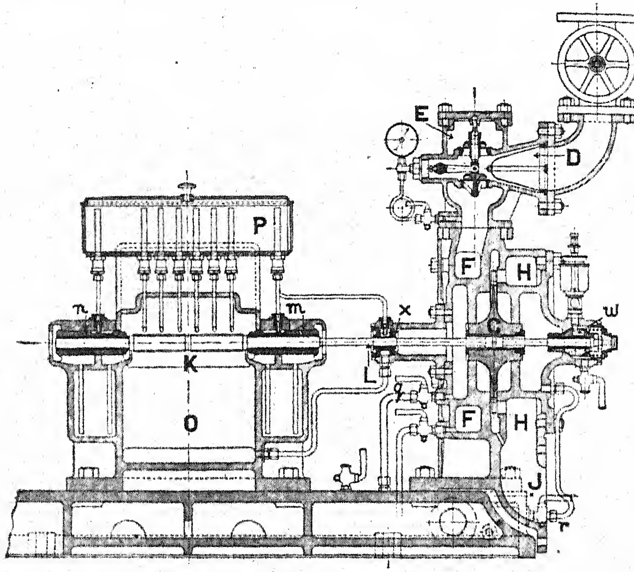


FIG. 181.

The small helical pinions K gear with large wheels on the driving shaft.

In order that the full advantage of the pressure drop may be obtained and also to direct the issuing steam in the required direction, a nozzle of the correct shape must be used. If a plain convergent nozzle is used, full expansion of the steam cannot be obtained if the pressure drop is large.

Suppose two vessels to be connected by a plain nozzle

and that each vessel contained steam at 150 lb. pressure per square inch ; then there would be no flow of steam from one vessel to the other, but suppose the steam pressure in one vessel (which may be assumed to be indefinitely large) is gradually reduced from 150 lb. pressure to 89 lb. pressure ; then it is found that the *weight* of steam discharged gradually increases (see Fig. 183). Any further reduction of pressure

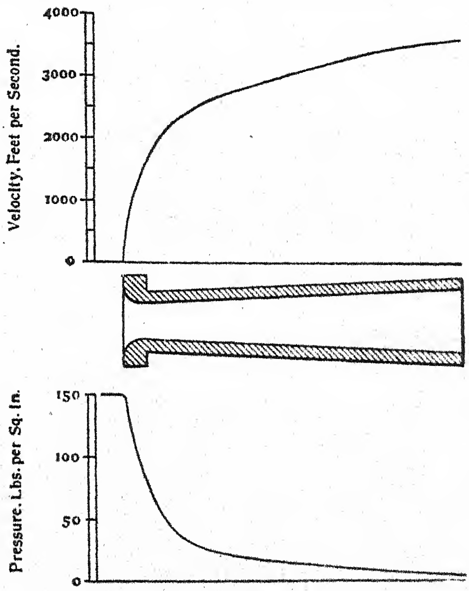


FIG. 182.

at the discharge end of the nozzle below 89 lb. does not affect the weight of steam discharged. It is found by experiment that the maximum weight of saturated steam passes through the nozzle when the discharge pressure is 58 per cent., or less than 58 per cent. of the initial pressure. This is illustrated by Fig. 183, which shows that the weight of steam discharged per second through a nozzle gradually increases with a falling back pressure and then remains stationary.

Actually the pressure at discharge with a plain nozzle is always 58 per cent. (54 per cent. with superheated steam) of the pressure at entry. If the pressure of the surroundings is less than this, the steam expands in all directions down to this pressure after leaving the nozzle, this energy being lost so far as useful work is concerned.

The area at the throat limits the amount of steam passing through the nozzle. If W=maximum weight of steam per

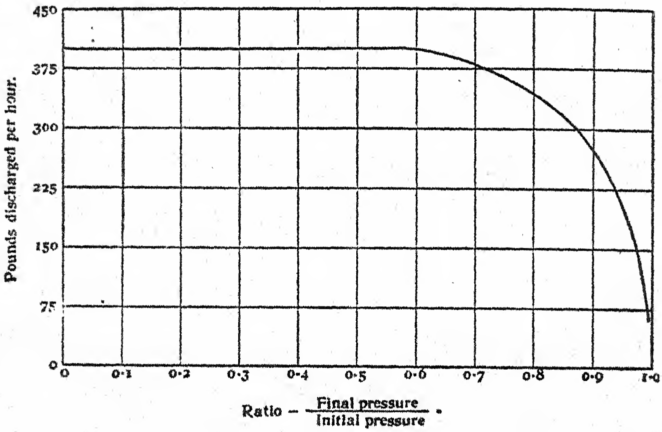


FIG. 183.

second ; a=area at the throat of the nozzle and P=absolute pressure in pounds per square inch, then $W=\frac{aP}{70}$. This formula is due to Napier, and applies to dry saturated steam only. In Fig. 183 the diameter at the throat has been taken to be a quarter of an inch and the initial pressure 150 lb. No losses have been considered.

It should be noticed that it is the *ratio* of back pressure to initial pressure that determines whether a plain nozzle is all that is necessary to convert the whole of the heat drop due to the drop of pressure into useful energy. For example, if the initial pressure is 160 lb. per sq. in. abs.

then the back pressure must not be less than $0{\cdot}58 \times 160 = 93$ lb. per sq. in., but if the initial pressure is 5 lb. per sq. in. the back pressure must not be less than 2·9 lb. per sq. in. In the first case a drop of pressure of 67 lb. per sq. in. is permissible, but in the second case a drop of pressure of 2·1 lb. per sq. in. only is permissible.

When the back pressure is too low for a plain nozzle, full expansion of the steam with the full final velocity due to the total pressure drop may be obtained by fitting an expanding (or divergent) mouthpiece to the orifice.

Fig. 182 shows the section of a De Laval nozzle with diagrams showing how the pressure falls as the steam passes through the nozzle and how the velocity increases, assuming ideal conditions.

Fig. 182 shows that the *velocity* of the steam constantly increases as the discharge pressure falls. The ratio of the area at the end of the nozzle to the area at the throat determines the pressure at the end of the nozzle, provided the steam is discharged into other steam at this pressure or at a lower pressure.

The steam is directed by the nozzle on to the turbine blades. These blades are specially curved to allow the steam to enter without shock. The *direction* of the steam is changed by the curved blades, and in thus changing the direction of the steam the blades receive their impulse.

Fig. 184 shows the blades of a De Laval turbine. The blades fit in the slots in the wheel, being fixed in position by a caulking tool.

The material of the blades for impulse turbines is required to be strong, smooth, durable, and non-corrosive.

For the highest efficiency of this turbine the wheel should have for the mean velocity of the rim, a speed equal to nearly half the velocity of the steam.

For a single-stage turbine of the De Laval type, where the whole of the expansion of the steam takes place in one stage, the best speed of the blading would be about 1800 ft. per second. Taking a mean diameter of 30 in. (circumference 7·85 ft.) the necessary speed of revolution would

thus be 13,800 r.p.m. At this speed the pull on a blade weighing 2 oz. due to centrifugal force alone would be 10,000 lb. (4·46 tons), which would be much too high for safety. In actual practice the speed in this case would be about 7500 r.p.m., which would reduce the pull to 1·32 tons.

The large number of revolutions of the De Laval turbine is one of its disadvantages, as these high speeds are not required in practice. It is usual to reduce the speed of the shaft which carries the driving pulley by spiral gearing, generally having a ratio of 10 to 1.

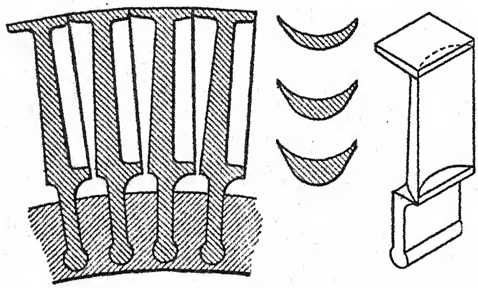


FIG. 184.

In the Rateau type of impulse turbine this high speed of rotation is avoided by using several impulse wheels placed in series and expanding the steam in stages.

The shaft and wheel of a De Laval turbine should be balanced as accurately as possible. Absolute accuracy is impossible, and the centre of gravity of the whole will not coincide *exactly* with the geometric axis. Thus when the wheel rotates, a force tending to bend the shaft is produced, which rapidly increases in amount until a certain speed is reached, called the 'critical speed.' If the turbine is run for a short time at this speed, the shaft would be broken. Above this speed the wheel will run steadily. The shaft is made flexible, so as to allow the shaft to rotate about its centre of gravity. The shaft will thus be slightly eccentric at full speed. The diameter of the shaft for a 300 H.P. turbine is only about 1¼ in.

The working speed of a De Laval turbine is always above the critical speed, and the vibration which might occur on passing the critical speed can be allowed for by having sufficient radial clearance. Radial clearance is unimportant in impulse turbines, as the pressure is the same on both sides of the wheel, and there is no tendency for the steam to leak past the wheel.

These turbines have been largely used for driving centrifugal pumps, ventilators, air compressors, and small electrical generators.

Losses in a De Laval Turbine.—The following approximate numbers will serve to show what the losses in a De Laval turbine are, where they occur, and some idea of their relative magnitude. Suppose the available energy obtained by expanding steam between two given pressures be represented by 100 when no losses take place. If the expansion takes place in a De Laval nozzle, there will be a loss of about 15 per cent. due to the friction of the steam against the sides of the nozzle and to the energy dissipated through the formation of eddies in the steam. This loss will, of course, be greater if the nozzle is not properly designed.

The friction between the steam and the blades causes a loss of about 10 per cent. The turbine wheel revolves in an atmosphere of steam, and the loss due to friction between the turbine disc and the steam is about 5 per cent. The energy left in the steam as it leaves the turbine is about 10 per cent. Radiation and mechanical friction together absorb about 5 per cent. The total loss is thus about 50 per cent.

Fig. 185 shows diagrammatically the fall of steam pressure in the nozzle of the De Laval turbine, and the corresponding increase of velocity of the steam before entering the wheel. It also shows the loss of velocity of the steam due to work done by it on the wheel blades, and the final condition of the steam on leaving the wheel both as to pressure and velocity.

Rateau Type Turbines.—Instead of expanding the steam from the boiler pressure to the condenser pressure in one

set of nozzles before entering the moving vanes as in the De Laval turbine (Fig. 185), the steam may be expanded in a series of stages—a small fall of pressure taking place at each stage. The fall of pressure occurs, however, not in the moving vanes themselves, but in the nozzles between the successive stages. This is the principle of the Rateau and Zoelly turbines, and is illustrated in Fig. 186. The steam after a small fall of pressure and a small increase of velocity, passes through a turbine wheel where its direction is changed and its absolute velocity is reduced. It next passes through a second row of nozzles, where its pressure falls further with a consequent increase of velocity, then through a second turbine wheel, and so on for several stages. A stage consists of one set of nozzles and one rotating wheel. The fall in pressure and the change of velocity, as the steam passes through the turbine, is shown by the pressure and velocity curves (Fig. 186). The low velocity of the steam due to dividing the fall of pressure into many stages enables the turbine to be run at a comparatively low rate of revolution, and to be coupled directly to electrical generators without the use of reduction gearing.

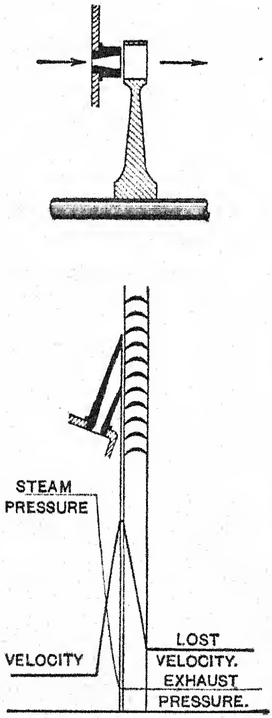


Fig. 185.

This type of turbine is *compounded for pressure*.

The main features of construction of the turbine are shown in section in Fig. 187, which shows a turbine consisting of eight impulse wheels.

Leakage of steam outwards and leakage of air inwards is prevented by having special glands G packed with carbon rings. A special water-sealed gland is used in later types for the low-pressure end.

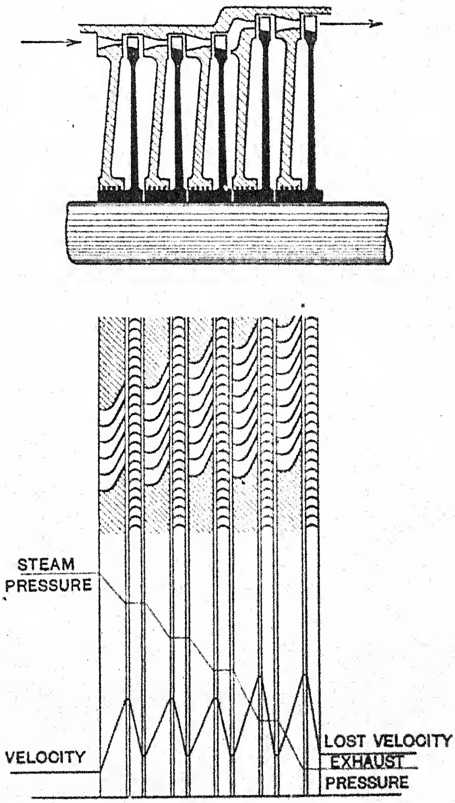


FIG. 186.

Curtis Turbine.—Another type of impulse turbine which can be run at a moderate speed and can be used for large powers is the Curtis turbine.

The principle of this type of turbine is shown in Fig. 188. Each impulse wheel has *two* or more rows of blades on its

circumference, and the casing has *one* or more rows of stationary blades for each wheel, the moving and stationary blades being placed alternately. The figure shows two rows of blades on each wheel and one row of stationary blades

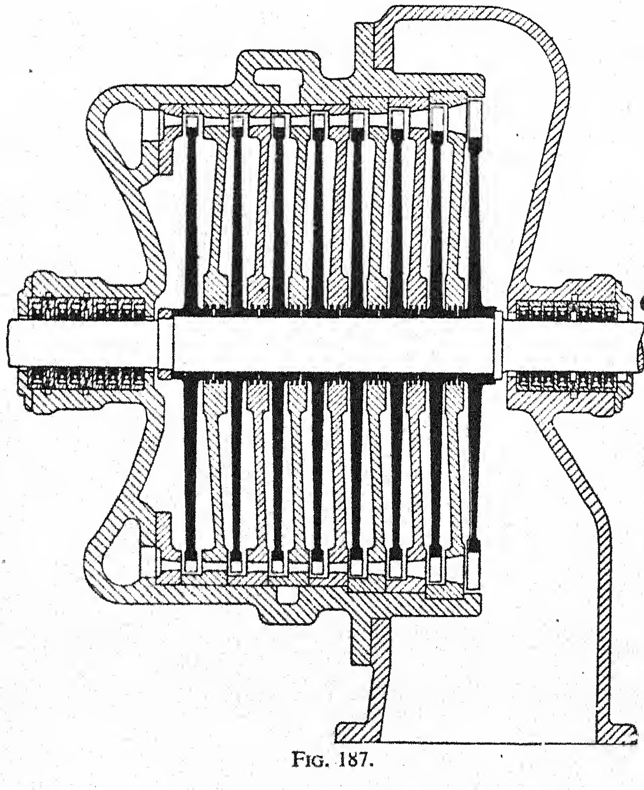


FIG. 187.

placed between the rows of rotating blades, and fixed to the casing. Each wheel is separated from the adjacent wheels by a diaphragm containing nozzles as in the Rateau type. It is thus compounded both for pressure and for velocity.

The steam expands in the first nozzle from, say, 160 lb.

per sq. in. to about 50 lb. per sq. in. There is no further fall of pressure as the steam passes through the moving and fixed blades (see pressure curves, Fig. 188). Hence there is no tendency for the steam to leak past the tips of the

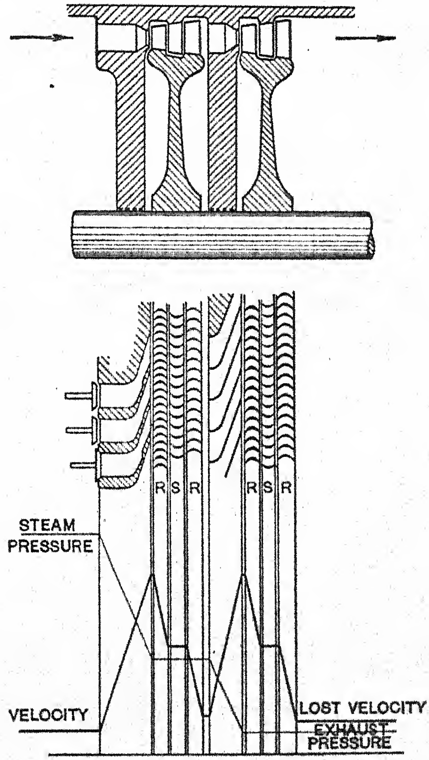


FIG. 188.

blades. The steam, after passing through the first row of moving blades and having lost some of its velocity, is directed by the fixed row of blades on to the second row of moving blades, where the greater portion of the remaining velocity is extracted.

Where two rows of blades are placed on the turbine

wheel, the velocity acquired by the steam during expansion in the nozzle is absorbed in two stages. This will allow of a much slower speed of the wheel, or if the wheel is run at a high speed, as is usual, it allows of a larger fall of pressure at one nozzle. This leads to the use of a smaller number of wheels and diaphragms. By using three rows of blades the number of wheels may be still further reduced, as a greater expansion will be possible at each row of nozzles. Unfortunately, the use of two rows of blades is less efficient than a single row, and three or more rows of blades give a still lower blade efficiency.

The large fall in pressure which takes place in the first nozzles of a Curtis turbine relieves the turbine casing from the boiler pressure, as the casing is only subjected to the pressure of the steam after expansion has taken place. A further advantage of the large fall in pressure outside the casing is that a lower pressure means a lower temperature of the steam, and the turbine casing is not subjected to the full temperature, due to the boiler pressure and superheat. This is important, as there is less danger of distortion of the casing and less danger from the growth or permanent expansion of the cast iron where cast iron is used for the casing. It has been found that cast iron will permanently expand if repeatedly heated to a high temperature, and even at temperatures as low as 550° F. In modern turbines the high pressure end of the casing is made of steel.

Fig. 189 is a section of an impulse turbine of 25,000 kilowatts capacity (35,000 B.H.P.) installed in 1920 in the Barton power station of the Manchester Corporation. It is designed for a steam pressure of 350 lb. per sq. in. gauge, steam temperature 700° F., and a vacuum of 29 in. (30 in. barometer). It has fourteen wheels, mean diameter at high pressure end 7 ft., tip diameter at low pressure end 9 ft., and the speed is 1500 r.p.m.

Fig. 190 is a section of an impulse turbine of 40,000 kilowatts capacity, installed ten years later in the same power station and working under the same steam conditions. The number of wheels in the high-pressure cylinder is

twenty-three and in the low-pressure cylinder eighteen. Mean diameter of high-pressure wheels 3·5 ft. Tip diameter of last wheel 10 ft. $2\frac{3}{4}$ in. Speed 1500 r.p.m.

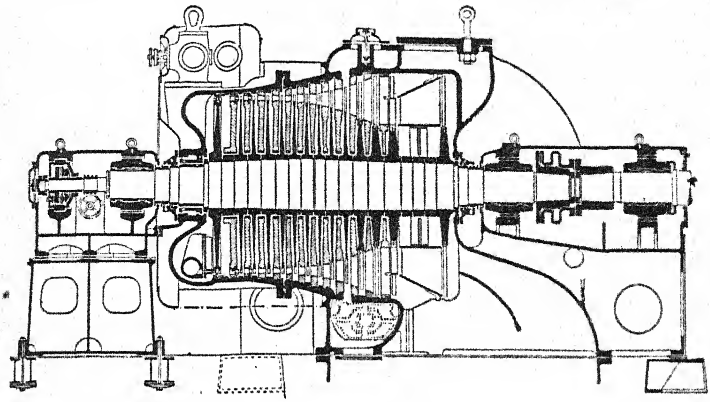


FIG. 189.

The thermal efficiency (steam to switchboard) of the latter turbine is 30·3 per cent. as compared with 26·8 per cent. for the older turbine, partly due to improvements in design and partly due to the use of a number of feed heaters

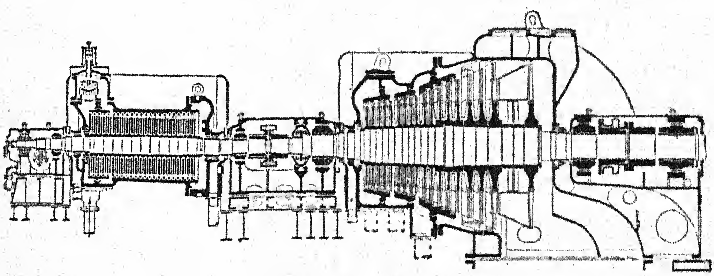


FIG. 190.

taking steam from a number of intermediate stages of the turbine. This amounts to a saving of 13 per cent. in fuel consumption with the newer turbine.

(Figs. 189 and 190 are reproduced by the courtesy of

Messrs. Metropolitan Vickers Electrical Co., Ltd., the manufacturers of the turbines, and of H. L. Guy, Esq., F.R.S., Chief Engineer of the Mechanical Department of the company.)

In all types of *impulse* turbine the pressure on the two sides of the rotating wheel is the same, and hence there is no tendency for the steam to leak over the tips of the blades. The clearance between the ends of the blades and the inside of the casing can thus be made ample, so as to prevent any danger of blade stripping, which might occur if the blades came into contact with the casing.

In impulse turbines, compounded for pressure, there is a difference of pressure on the two sides of the diaphragm containing the nozzle, and hence a tendency for steam to leak from one side of the diaphragm to the other at the point where the shaft passes through the stationary diaphragm. The area for possible leakage at this point is, however, comparatively small.

Parsons Turbine.—Fig. 191 shows a section of a Parsons turbine of a simple type.

It consists of a rotor to which rings of blades are attached, and a casing having rings of stationary blades attached internally. The rotor is supported at each end by bearings, and the casing consists of two parts bolted together.

The steam enters at A, at the boiler pressure, and is usually superheated. It passes alternately through the stationary and moving blades. There is a continuous fall of pressure as the steam passes through the turbine, with a consequent increase of volume. It is obvious that if the steam velocity through the turbine is to be constant the area through which the steam passes must be increased as the volume increases. If the blades are all of the same shape, and the same number is used per ring, then the blades in each successive ring must be lengthened. For convenience of construction it is usual to have several rings of blades of the same height and spacing, and to increase the diameter of the rotor at intervals. At the exhaust end the volume of the steam rapidly increases as the pressure

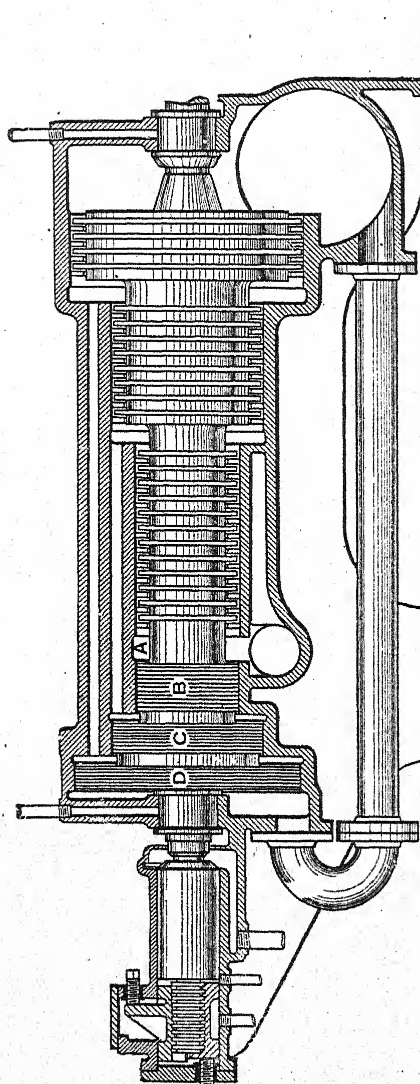


FIG. 191.

falls, and the blades whilst retaining the same height may be spaced farther apart, and also be made with a flatter curvature. The following example will illustrate why a change of area per stage is more necessary at the low-pressure end than at the high-pressure end. Suppose the pressure falls in the first five stages (a stage consists of one rotating and one stationary set of blades) from 150 lb. to 140 lb.; the volume will increase from 3·01 to 3·20 cu. ft. per lb., assuming adiabatic expansion of the steam. This is an increase of about 6 per cent. If the pressure falls in the low-pressure stage from 4 lb. to 3 lb. per sq. in., the volume will increase from 90 cu. ft. per lb. to 116 cu. ft., allowing for wetness, due to expansion, or an increase of about 29 per cent.

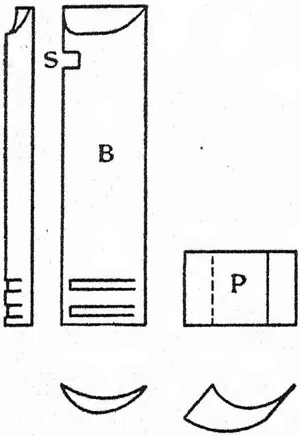


FIG. 192.

The cross-sectional area between the blades should be correctly designed, so as to allow for a small fall of pressure and an increased velocity of the steam.

The steam after passing through the turbine escapes to the condenser.

Turbine Blades.—The blades have been made of brass, gunmetal, steel, and an alloy of 80 per cent. copper with 20 per cent. nickel. Fig. 192 illustrates a blade, B, and packing piece, P, of a Parsons marine turbine. The slot S near the top of the blade is for the binding wire, which passes from blade to blade, and is wired or soldered to each blade. The blades are often thinned at their lips to about $\frac{1}{64}$ in. to prevent serious damage, if they should come in contact with the inside of the casing. They are secured in position by the packing piece, which is placed between each pair of blades, and caulked. The two small grooves

at the bottom of the blade assist in holding the blade securely.

The pressure of the steam on the blades produces an unbalanced axial pressure on the rotor, tending to force the rotor in the direction of the steam exit. To balance this axial pressure three dummy pistons, B, C, and D, are introduced, Fig. 191, each of which is connected with one of the sections. The pressure on any one of the dummy or balance pistons is the same as that in the section to which it is connected, but in an opposite direction.

To prevent leakage of steam past the balance pistons, a large number of grooves are turned in their circumference. This is found to be very effective in preventing leakage. The pressure on the outside of the large piston is maintained at the condenser pressure by connecting the space in which the outside face rotates with the exhaust.

Care is required in the *axial* adjustment of the rotating blades to prevent them from coming in contact with the stationary blades. This is accomplished by having a thrust-block at E. A number of grooves are turned in the shaft, and a number of rings in the bearing project into these grooves.

The upper and lower halves of the bearing are adjustable separately by micrometer screws with dials to show the exact position of the blades. If the upper half of the bearing tends to press the shaft to the right, then the lower half is adjusted to tend to press the shaft to the left. In this way the rotor is fixed in a definite position and the axial clearance is maintained.

The *radial* clearance is made as small as possible, as a high efficiency of the turbine depends largely on a small radial clearance.

The pressure on the two sides of the moving blade being different, a large clearance causes considerable leakage of steam past the blades (see Fig. 193, in which C_1 represents the clearance (not to scale) between the moving blades and casing, and C_2 the clearance between the rotor and the stationary blades).

The actual clearance of a 48-in. rotor is about 0·048 in., or about $\frac{1}{1000}$ in. per inch of diameter.

The bearings are lubricated by oil which is forced in at a pressure of from 5 to 10 lb. per sq. in. The bearings are usually of white metal, with a safety brass bearing slightly below the level of the white metal. This safety bearing will support the rotor and prevent the blades coming in contact with the casing, in the event of the white metal becoming overheated and being melted out of the bearing.

The maintenance of a continuous flow of oil to the turbine bearings is of the utmost importance, in order to prevent the bearings from becoming overheated. The oil from large bearings should be cooled before returning it to the bearings.

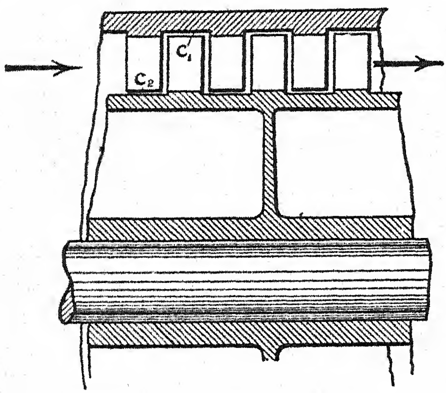


FIG. 193.

The principle of design adopted in the Parsons type of turbine is shown in Fig. 194. There is a small fall of pressure in both the fixed and the moving blades, as shown by the pressure and velocity curves (Fig. 194). The pressure fall is less in each stage than in the impulse type, and hence more stages are necessary. The absolute velocity developed by the expansion of the steam during fall of pressure is never very high, and the peripheral speed of the turbine is sufficiently low to allow of its being coupled direct to a generator.*

In addition to the types of turbine already described,

* For particulars of modern turbine of the Parsons type see "Ripper's Steam Engine Theory and Practice."

turbines are constructed which combine two types in one turbine. A common combination is to use the Curtis principle at the high-pressure end and the Rateau principle

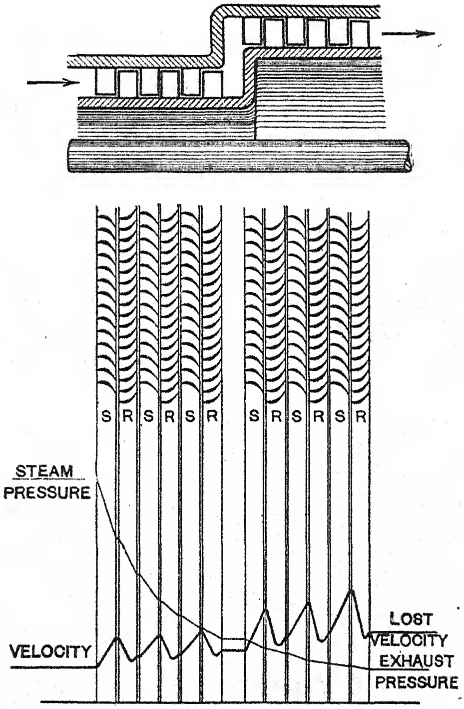


FIG. 194.

for the low-pressure end. This type of turbine is called a Curtis-Rateau turbine.

Curtis-Rateau Turbine.—The advantage of using the Rateau principle for the low-pressure end is that it is rather more efficient than the Curtis principle, and where fuel is expensive the use of the more expensive Rateau wheels will be justified. The use of the Curtis principle at the

high-pressure end reduces the number of wheels and shortens the turbine shaft. The Curtis and Rateau types have nearly similar efficiencies at the high-pressure end.

Another combination is to use the Curtis wheel for the high-pressure end and the Parsons drum for the low-pressure end.

Curtis-Parsons Turbine.—The efficiency of the Parsons reaction turbine is lowest at the high-pressure and highest at the low-pressure end. The short blades at the high-pressure end of a Parsons turbine give rise to leakage and eddies, and hence this portion of the turbine is somewhat less efficient The introduction of a Curtis wheel shortens the turbine very considerably by reducing the number of rows of blades, and at the same time it gives a good efficiency. Fig. 195 shows a section of a Curtis-Parsons turbine. The steam enters as shown, and after expanding in the nozzles passes through the Curtis impulse wheel, and then passes through the reaction blading on the Parsons drum. The steam expands to about 50 lb. per sq. in. in the nozzles of the impulse wheel, and the remainder of the expansion takes place in the reaction blading.

Leakage of steam outwards and the leakage of air inwards is prevented by having special glands G. Where the pressure inside the turbine is less than that of the atmosphere, air tends to leak into the turbine and reduce the vacuum. The glands are packed with steam to prevent this, and any leakage inwards will be steam, which will have little effect on the vacuum.

With turbines of 50,000 H.P. and upward the blade height at the high-pressure end is sufficient to dispense with the Curtis wheel and so obtain a somewhat higher efficiency, an increase of only 1 per cent. representing a considerable saving in fuel cost in the course of a working year.

Example.—A turbine of 50,000 H.P. uses 1 lb. of coal per hour per horsepower. Calculate the annual saving in fuel consumption due

to a reduction in steam consumption of 1 per cent. if the turbine works at full load for 4000 hours per annum.

$$\text{Saving per hour} = \frac{50{,}000}{100} = 500 \text{ lb.}$$

Saving per annum $= 500 \times 4000 = 2{,}000{,}000$ lb. $= 892$ tons.

If the fuel costs 25*s*. per ton, the annual saving will be £1116.

Velocity of Flow from a Nozzle.—The velocity of flow from a nozzle can be calculated accurately with the aid of

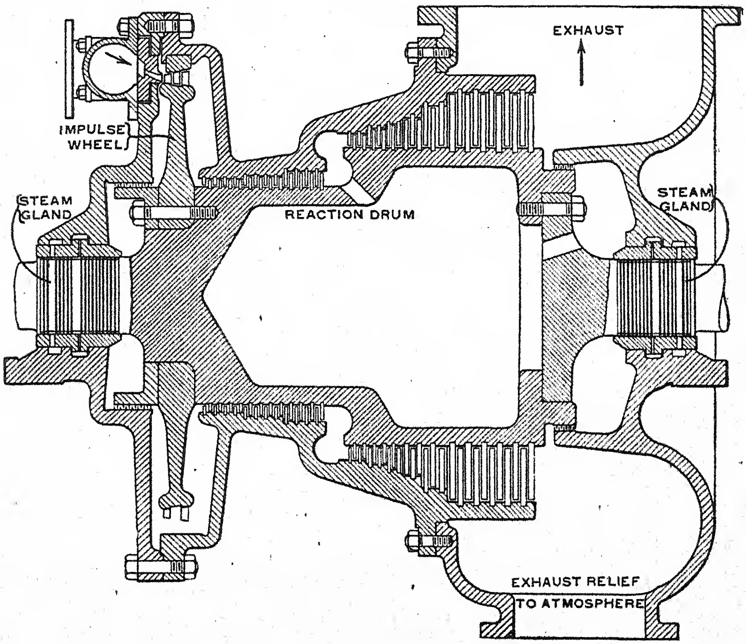


FIG. 195.

entropy diagrams, consideration of which is beyond the scope of this book. The difference between the total heats per pound at entrance and exit (known as the *heat drop*) is the heat converted to kinetic energy. The total heat per pound at entry will be known, but the total heat at exit will depend upon the wetness of the steam, due to the transformation of heat into velocity energy.

A formula which will give a roughly approximate value for the velocity of exit from a properly shaped nozzle (neglecting friction losses) is

$$V=100\sqrt{p_1 v_1 \,.\, \text{hyp. log.} \frac{p_1}{p_2}}$$

where V=velocity of issue (ft. per second)

p_1=initial pressure (lb. per sq. in. abs.)

p_2=pressure at exit (lb. per sq. in. abs.)

v_1=specific volume at entry (cu. ft. per lb.)

Example.—$p_1=200$ lb. per sq. in. (dry saturated) ; $v_1=2{\cdot}29$ (from Table III) ;

$p_2=120$ lb. per sq. in.

$$\text{hyp. log.} \frac{p_1}{p_2}=\text{hyp. log. } 1{\cdot}67=0{\cdot}513$$

$$V=100\sqrt{200\times 2{\cdot}29\times 0{\cdot}513}=1535 \text{ ft. per sec.}$$

The value by a correct method, neglecting losses, is 1480 ft. per sec.

Velocity Diagrams.—These diagrams are essential aids to the correct design of steam turbines. It is necessary, however, to have a clear idea of the terms absolute and relative velocity before proceeding. *Absolute* velocity is the velocity as compared with stationary objects. *Relative* velocity is the velocity in relation to that of a moving body.

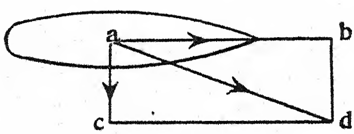


FIG. 196.

Suppose a ship is travelling at the rate of 20 ft. per second, and a ball be thrown from the ship at right angles to the ship at 15 ft. per second, then relative to the ship the velocity of the ball will be 15 ft. per second at right angles to the ship. The absolute velocity relative to the earth is obtained by constructing a parallelogram as in Fig. 196, in which *ab* represents the ship's velocity and *ac* the velocity of the ball. The absolute velocity and direction of the ball will be *ad*.

Let the velocity of the jet leaving a De Laval nozzle be V_1 ft. per second and make an angle with the wheel of 20° ; let the velocity of the wheel be V ft. per second. Find the

relative velocity of the steam and the correct entrance angle of the blade so that the steam may enter without shock. Construct the diagram of velocities by drawing $AB=V_1$, and making an angle of 20° with the plane of the wheel DB.

Make $BC=V$ the velocity of the wheel. Then AC is the relative velocity and direction of the steam with respect to the wheel. The angle ACD is the entrance angle for the blades. The exit angle is approximately the same for a De Laval turbine.

There is no change of velocity in passing through the turbine, and therefore AC represents the velocity on leaving relative to the blade. If $EF=AC$ represents the magnitude and direction of the velocity relative to the blade on leaving, then, if the diagram of velocities is constructed so that $FG=V$, the velocity of the wheel, EG is the absolute velocity of the steam on leaving.

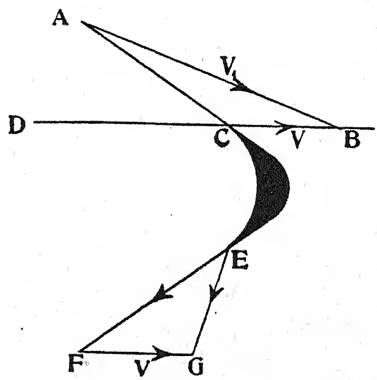


FIG. 197.

By making the inlet and outlet angles equal there is practically no end thrust on the spindle, since the axial velocity at exit is the same as at entry and the pressure also is the same on each side of the wheel.

It will be noticed from Fig. 197 that the *back* of the blade is made tangential to AC. The front of the blade is so shaped that the area for passage of steam remains constant from the entrance to exit.

In the *reaction* turbine the blading is made unsymmetrical so that the area at exit is *less* than the area at entrance (measured in the direction of flow). Since the volume leaving is nearly the same as the volume entering, the

velocity at exit is greater than the velocity at entry, this giving the reaction effect.

As a heat engine the principal merit of the steam turbine lies in its ability to make available for useful work the energy in that portion of the steam which has hitherto passed away to waste in the exhaust of the reciprocating engine.

Thus in Fig. 198, if pressure *ed* represents the pressure at exhaust in the reciprocating engine, beyond which it would not be profitable to expand, it is possible to obtain the still further area *edfg* by continued expansion of the steam in the turbine. And this is further increased by the

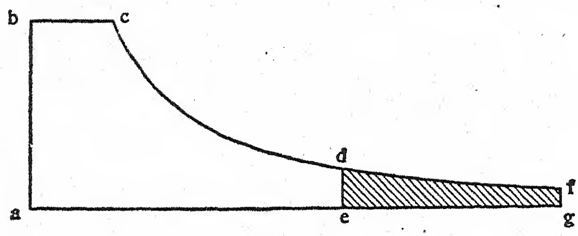


FIG. 198.

superior vacuum obtained in turbine plants, thereby giving a lower back-pressure line *ag*.

Fig. 198 is not to scale. Actually *eg* is considerably longer than *ae*, as the steam in a modern turbine is expanded practically down to back pressure. The advantage of a decrease of back pressure is therefore considerably greater for a turbine than for a reciprocating engine.

In the steam turbine it is found that an increase of vacuum from 25 to 26 in. improves the economy about 4 per cent. ; an increase from 26 to 27 in. gives about 5 per cent. more economy ; and a 28-in. vacuum is about 6 per cent. better than a 27-in. vacuum. An improvement of about 15 per cent. is obtained by increasing the vacuum from 25 to 28 in.

In modern practice with large turbines a vacuum of over 29 in. (barometer at 30 in.) is maintained, giving still

further economy in fuel consumption. At so low an exhaust pressure the volume of steam to be passed is very large (about 600 cu. ft. per lb.) and various devices such as double flow ends are used to avoid excessive lengths of blading at the low-pressure end (see Figs. 189 and 190).

Exhaust Steam Turbines.—The great economy of the turbine at low pressures has led to the use of the exhaust steam turbine to take the steam leaving the reciprocating engines at atmospheric pressure.

The clearances in these turbines do not require to be as fine as when high pressures are used, and the temperatures being low there is little distortion of the casing. An economy of about 20 per cent. is possible by adding an exhaust turbine in this way.

Comparison of Turbine Types.—*Impulse Turbines.*

1. Best speed of blading is about 45 per cent. of the steam speed.
2. Maximum tip speed of blading (limited by centrifugal stresses) is 750–1000 ft. per second, depending on material of disc and blading.
3. Partial admission possible at entry. Thus the blade speed can be settled without reference to least permissible blade height.
4. No leakage over tips of blades. Fine tip clearances unnecessary.
5. High superheat temperatures reduced in nozzles before reaching blading.
6. Blading losses higher than in reaction turbine, owing to higher steam speeds.
7. Losses due to skin friction of discs.
8. Leakage losses past diaphragm at shaft. These are small.
9. No appreciable end thrust.
10. Possibilities of ‘ whirling ’ of shaft.
11. Few stages.

Reaction Turbines.

1. Best blading speed about 92 per cent. of steam speed.
2. Maximum permissible blading speed for a drum is

about 60 per cent. of that for a disc. In very large reaction turbines, however, the low-pressure rotor is built up of a number of discs in close contact.

3. All blades must run full of steam, so that at the high-pressure end the blade circumference must be small enough to give reasonable height.

4. Owing to possible leakage over blade tips very small radial clearance is necessary. The alternative is 'axial' clearance.

5. Reduced blading losses owing to lower steam speeds.

6. Skin friction of drum is less than with disc, owing to smaller exposed area.

7. Provision for end thrust is necessary, by 'dummy' pistons or otherwise.

8. Leakage losses past tips of blading and past dummies.

9. Many stages.

10. A drum, being stiffer, is less likely to whirl than a loaded shaft.

CHAPTER XIX

PROPERTIES OF GASES

Let a portion of gas be introduced into a cylinder which is closed at one end and fitted with a movable piston. Then the gas will fill every part of the space beneath the piston, and exert a uniform pressure on each square inch of surface with which it is in contact. If the internal volume of the cylinder be increased, by lifting the piston, the gas will still completely fill the space, but it will be less dense—that is, it will weigh less per cubic foot—and it will exert less pressure per square inch of surface with which it is in contact.

If the gas be compressed into a smaller space, it will become more dense, and it will exert a greater pressure per square inch.

The relation between the volume and pressure of a perfect gas at constant temperature is expressed by the following law, known as Boyle's Law :

'The volume of a given weight of gas varies inversely as the pressure, the temperature remaining constant.'

This may be illustrated as follows :

Let a cylinder (Fig. 199) be closed at one end and contain a movable piston, and let the piston, when in position *a*, enclose 1 cu. ft. of gas under, say, 15 lb. per sq. in.

Suppose, now, that weights be added to the piston till the pressure on the enclosed gas is equal to 30 lb. per sq. in. Then, by the law just stated, the pressure on the gas being doubled, the volume will be reduced one-half. Hence the piston now occupies position *b*, so that $eb=\frac{1}{2}ea$.

Again, apply to the piston a pressure equal to 60 lb. per sq. in., or four atmospheres. The pressure on the gas being now four times the original pressure, its volume is one-fourth of its original volume, and the piston now falls

to c, so that $ec=\frac{1}{4}ea$. Again, apply to the piston a pressure equal to 120 lb. per sq. in., or eight atmospheres. The pressure on the gas being eight times the original pressure, the volume is now one-eighth of the original volume, and the piston falls to d, so that $ed=\frac{1}{8}ea$. If now horizontals be drawn from the respective piston positions the length of which is equal to the pressure at these positions to any scale, and a curve be drawn through the extremities of the lines, the student will recognise the curve as being similar to that of a steam engine hypothetical indicator diagram if the book be held so that the cylinder is horizontal. This curve is called a rectangular hyperbola.

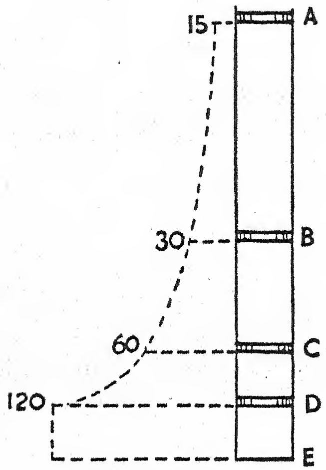


FIG. 199.

Boyle's Law may also be expressed thus :

If V is the volume at pressure P			
then $\frac{1}{2}$V	,,	,,	2P
$\frac{1}{3}$V	,,	,,	3P
$\frac{1}{4}$V	,,	,,	4P
and so on ; or,			
2V	,,	,,	$\frac{1}{2}$P
3V	,,	,,	$\frac{1}{3}$P
4V	,,	,,	$\frac{1}{4}$P

From which it is evident that in each case, if the pressure be multiplied by the volume, the result is a constant number.

Boyle's Law is only true for gases and only if the temperature remains constant while the pressure and volume are changed.

The *Law of Charles* states that 'the volume of a given weight of gas at constant pressure, or the pressure of the gas at constant volume, is proportional to the *absolute* temperature of the gas.'

The two laws can be expressed by the equation

$$\frac{p.v}{T}=\text{constant}$$

From this equation we see that if T is constant, $p.v$ =constant (Boyle's Law). If v is constant, p is proportional to T, and if p is constant, v is proportional to T (Charles's Law).

The above equation is a very important one where calculations in connection with internal combustion engines have to be made, and expresses the fact that if we take a certain quantity of gas the value of $p.v/T$ remains the same, however we alter its pressure, volume, or temperature.

(NOTE.—p and T must be in *absolute* units.)

Example 1.—A quantity of air in a cylinder under a movable piston occupies 10 cu. ft. at 60° F.; what volume will it occupy if heated to 250° F. under the same constant pressure?

Here, the volume occupied by the air will evidently be greater, and in proportion to the absolute temperature, thus:

60° F.= 60+460=520 absolute temperature
250° F.=250+460=710 „ „

$$\text{Then, volume at } 250° \text{ F.} = \text{volume at } 60° \times \frac{710}{520}$$
$$= 10 \times \frac{710}{520}$$
$$= 13{\cdot}65 \text{ cu. ft.}$$

Example 2.—A volume of air at 212° F. is confined in a rigid cylindrical vessel, and exerts a pressure of 15 lb. per sq. in.; find the pressure exerted by the air when the temperature is increased to 300° F., the volume, of course, remaining the same.

Here, by the above law, the pressure exerted by the air will be greater, and in proportion to the absolute temperature; then,

212° F.=212+461=673 absolute
300° F.=300+461=761 „

$$\text{Then pressure at } 300° \text{ F.} = \text{pressure at } 212 \times \frac{761}{673}$$
$$= 15 \times \frac{761}{673}$$
$$= 16{\cdot}96 \text{ lb. per sq. in.}$$

If the temperatures are given in degrees Centigrade instead of Fahrenheit, then to find the absolute temperature add 273, thus :

Example 3.—A certain quantity of gas occupies 20 cu. ft. at 15° C.; what volume will it occupy if its temperature is raised to 100° C., the pressure on the gas remaining constant ?

$$15^\circ \text{ C.} = 15+273=288 \text{ absolute}$$
$$100^\circ \text{ C.}=100+273=373 \quad \text{,,}$$
$$\text{then } 20\times\frac{373}{288}=25{\cdot}9 \text{ cu. ft.}$$

Example 4.—A volume of 5 cu. ft. of gas at 70° C. and 14 lb. per sq. in. is compressed to a volume of 0·4 cu. ft. If its final pressure is 250 lb. per sq. in., what is its final temperature ?

Let **T**=final temperature (° C. absolute)

then
$$\frac{14\times5}{70+273}=\frac{250\times0{\cdot}4}{T}$$
$$T=\frac{250\times0{\cdot}4\times343}{14\times5}=490^\circ \text{ C. abs.}$$
$$=217^\circ \text{ C.}$$

Specific Heats of a Gas.—The specific heat of a substance has been defined (see p. 8) as the number of thermal units required to raise the temperature of one pound of a substance by one degree. In the case of a solid the amount of expansion that occurs is so small that the work done in expanding against the pressure of the surrounding air is quite negligible. In the case of a gas, however, if it is allowed to expand when heated an appreciable amount of work may be done in expanding and more heat is required to raise its temperature than if it is confined so that it cannot expand.

We thus have two specific heats for a gas : (1) the specific heat at constant volume, and (2) the specific heat at constant pressure.

The *specific heat at constant volume* is the number of thermal units required to raise the temperature of one pound of the gas by one degree, the gas being confined so that its volume remains constant during heating.

The *specific heat at constant pressure* is the number of thermal units required to raise the temperature of one pound of the gas by one degree, the gas being allowed to expand during heating so that its pressure remains constant.

The work (in foot-pounds) done by a gas in expanding (pressure constant) is given by

Pressure (lb. per sq. ft.) × change of volume (cu. ft.)

No matter how the pressure alters, no work is done unless a change of volume occurs. For heating a gas at constant pressure extra heat is required for the work done in expansion over and above that required to increase its temperature (i.e. its internal energy), so that the specific heat at constant pressure (C_p) is always greater than the specific heat at constant volume (C_v).

Values of the specific heats (at ordinary temperatures) of some of the more common gases are given below :

Gas	C_p	C_v	C_p/C_v	C_p-C_v
Air	0·24	0·17	1·41	0·07
CO_2	0·20	0·153	1·31	0·047
Oxygen	0·22	0·157	1·40	0·063
Nitrogen	0·25	0·176	1·41	0·074

If we take *one pound* of a gas, the relation between its pressure, volume, and absolute temperature may be shown from theoretical considerations to be

$$p.v.=J.(C_p-C_v).T$$

where p=pressure in pounds per square foot (absolute)
v=volume in cubic feet
J=Joule's equivalent

Example 5.—Find the volume of 1 lb. of air at 50 lb. per sq. in. and 160° C.

$$p=50\times144\ ;\ T=160+273=433\ ;\ J=1400\ ;\ C_p-C_v=0{\cdot}07$$
$$\therefore\ 50\times144\times v=1400\times0{\cdot}07\times433$$
$$\therefore\ v=5{\cdot}89 \text{ cu. ft.}$$

Example 6.—Find the weight of 40 cu. ft. of oxygen at 600 lb. per sq. in. and 75° C.

Let v=volume of 1 lb. at above pressure and temperature ; $p=600\times144$; $T=348$; $J=1400$; $C_p-C_v=0{\cdot}063$.

$$\therefore\ 600\times144\times v=1400\times0{\cdot}063\times348$$
$$\therefore\ v=0{\cdot}356 \text{ cu. ft. per lb.}$$
$$\therefore \text{ Weight of 40 cu. ft.}=\frac{40}{0{\cdot}356}=112 \text{ lb.}$$

CHAPTER XX

INTERNAL COMBUSTION ENGINES

In the case of steam engines and turbines the combustion of the fuel, which supplies the heat, part of which is converted into work in the engine, takes place before the working fluid (steam) is admitted to the engine cylinder. In internal combustion engines, as the name implies, combustion of the fuel takes place *inside* the cylinder.

Internal combustion engines may be classified according to the nature of the fuel used, as (1) gas engines, (2) petrol engines, (3) oil engines.

In the gas engine the fuel (coal gas or producer gas) is supplied to the cylinder mixed with the necessary amount of air, and the mixture is ignited by means of a spark.

In the petrol engine the fuel (petrol or mixtures of petrol with benzol or alcohol) is first vaporised after passing through some form of carburetter and mixed with air before ignition is effected by means of a spark.

In the modern oil engine the heavy oil used cannot easily be vaporised before admission to the cylinder. It is therefore introduced in a very fine spray, mixed with air and ignited by the heat of compression, which is necessarily very much higher than the compression pressures used in gas and petrol engines.

Any of the above types may work on what is known as the four-stroke cycle or on the two-stroke cycle. The four-stroke cycle system was advocated by Beau de Rochas in 1862 and introduced by Otto in 1876, and is usually called the Otto cycle. It requires four strokes of the piston to complete the cycle of operations in the cylinder.

1*st Stroke* (*Outstroke*).—A charge of gas and air is taken into the cylinder. Since the pressure inside the cylinder must be less than the pressure outside if flow is to take

place the pressure in the cylinder will be less than atmospheric during this stroke (AB, Fig. 207). The fall of pressure depends upon the speed and the area of opening of the gas and air valves.

2nd Stroke (*Instroke*).—All valves closed. The charge is compressed on the return of the piston (BC, Fig. 206).

3rd Stroke (*Outstroke*).—All valves closed. The charge is fired at or before the commencement of this stroke, and the pressure rises due to the heat generated by combustion (CD, Fig. 206). The gases now expand and do work on the piston (DE, Fig. 206). Near the end of the stroke (E, Fig. 206) the exhaust valve opens and the pressure falls.

4th Stroke (*Instroke*).—The piston expels the exhaust gases from the cylinder. Since the pressure inside the cylinder must be greater than the pressure outside if flow is to take place, the pressure in the cylinder during exhaust will be greater than atmospheric (FA, Fig. 207). This increase of pressure depends upon the length and size of the exhaust pipe and the resistance offered by the silencer and exhaust valve.

The four strokes are illustrated by Fig. 200.

Fig. 201 shows the *approximate* positions of the crank when the various operations occur in a *four*-stroke cycle gas or oil engine. The positions of the crank coinciding with these operations vary with the speed and size of the engine.

It will be noticed that the gas and air valves close after the crank has passed the dead centre. By this means a stronger charge is detained. This is possible because the pressure of the gas and air in the cylinder during the suction stroke is less than the atmospheric pressure, and the piston must compress the gas slightly before atmospheric pressure is attained. Also the velocity of the gas through the valves having some momentum is not quickly reversed in direction.

The Two-stroke Cycle.—In the two-stroke cycle it is necessary to provide for the various operations previously explained in two strokes instead of in four. In order to accomplish this a mixture of the working gases must be

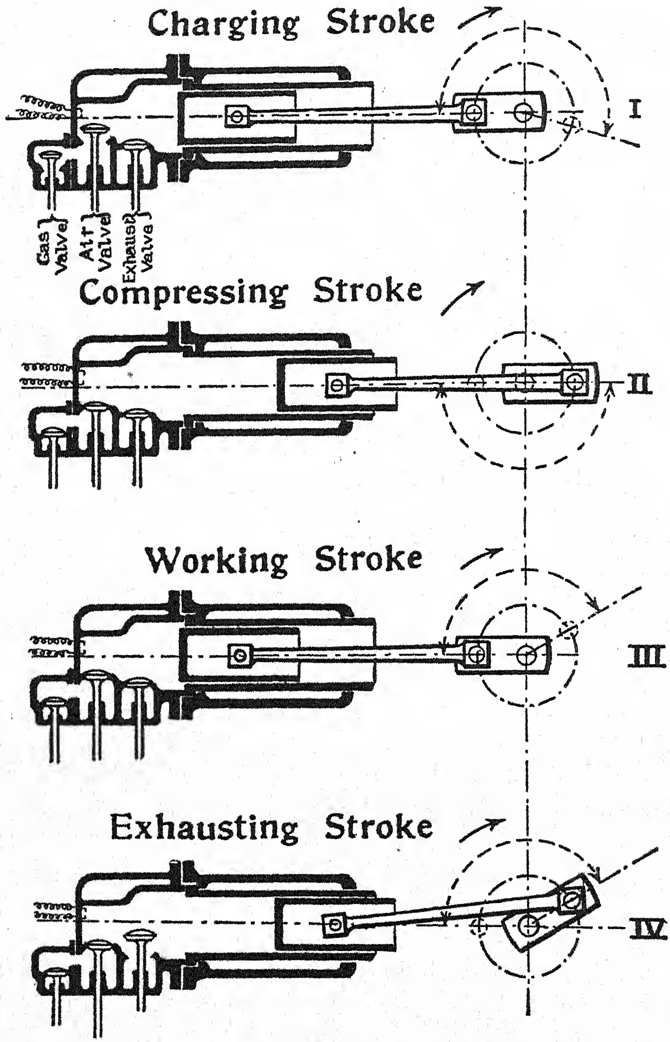

Charging Stroke
I
Gas Valve
Air Valve
Exhaust Valve
Compressing Stroke
II
Working Stroke
III
Exhausting Stroke
IV

FIG. 200.

supplied to the cylinder under pressure. In larger types of engines this is secured by using a separate pump or pumps. In the motor cycle two-stroke engine the pressure of the new charge is secured by an ingenious method of utilising the enclosed crank case, into which the working charge is first drawn by the motion of the piston on its inward or upward stroke and then compressed by the motion of the piston on its downward or outward stroke. The supply of the working mixture under pressure serves two purposes. First it secures its ready inflow into the cylinder, and

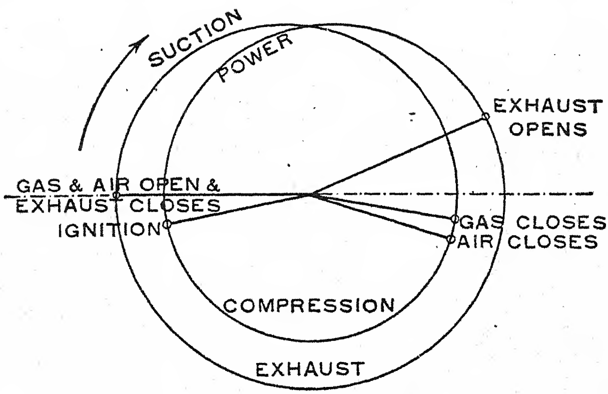


FIG. 201.

secondly causes the inflowing gas to sweep out the exhaust gases which it is intended to displace, an operation known as scavenging. A very simple case of this arrangement is illustrated in the two-stroke motor cycle engine shown in Figs. 202, 203 and 204.

1*st Stroke* (Fig. A).—A newly ignited charge is exerting pressure on the top of the piston, while fresh mixture from the carburettor is passing into the closed crank case by way of the open port I. (Fig. B) The charge which was ignited at the beginning of the stroke, having expanded and driven the piston down, is leaving the cylinder and passing into the air through the exhaust port E, now partly uncovered

by the piston. The mixture in the crank case has been slightly compressed by the descending piston in readiness to pass through the transfer port G as soon as it opens. The point of opening occurs when the piston is on its downward stroke and when the crank is situated about 30° or more from the outer dead centre.

2nd Stroke (Fig. C).—The piston is shown at the begin-

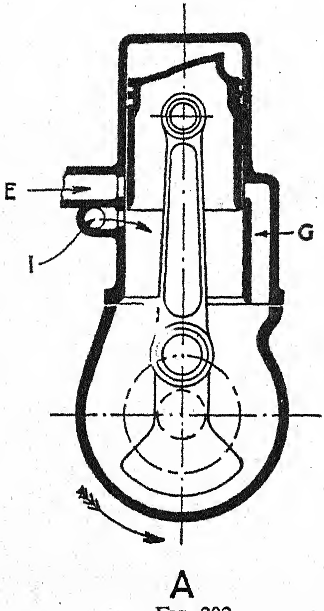


FIG. 202.

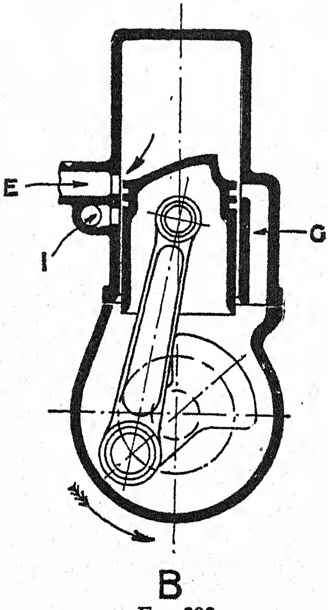


FIG. 203.

ning of the return or upward stroke ; the exhaust port and the transfer port are fully open. The compressed mixture from the crank case is passing through the transfer port into the cylinder, where it is deflected upwards by the special form of the piston to aid in sweeping the burnt gas from the cylinder. The fresh charge passes into the cylinder while the crank moves from about 30° before to 30° past the outer dead centre. (Fig. D) The piston covers all the

ports ; compression takes place above the ascending piston during the remainder of the upward stroke, and a partial vacuum is formed in the crank chamber. The end of this stroke completes the cycle.

This type of two-stroke engine has disadvantages as compared with the four-stroke engine for gas and petrol engines. In the first place it is almost impossible to obtain

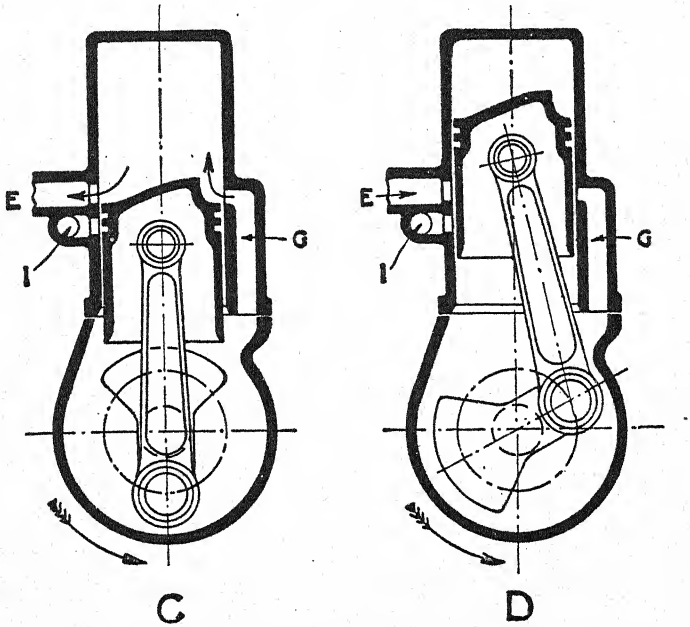


FIG. 204.

reasonably efficient scavenging of the exhaust gases without passing an appreciable portion of the incoming charge through the exhaust port, so that the fuel consumption per horsepower-hour is higher than that of a four-stroke engine. This difficulty is intensified at high speeds and at very low speeds. The mean effective pressure seldom exceeds 60 lb. per sq. in. at full load, while a mean effective

pressure of at least 140 lb. per sq. in. is quite practicable with a four-stroke petrol engine, so that little, if any, increase of power is obtained by using a two-stroke engine. It must also be remembered that nearly double the amount of heat is generated in a given time, so that cooling problems are intensified. The reciprocating parts are also heavier, which limits the maximum speed of revolution. For these reasons this type of two-stroke engine is now rarely used for either gas or petrol engines.

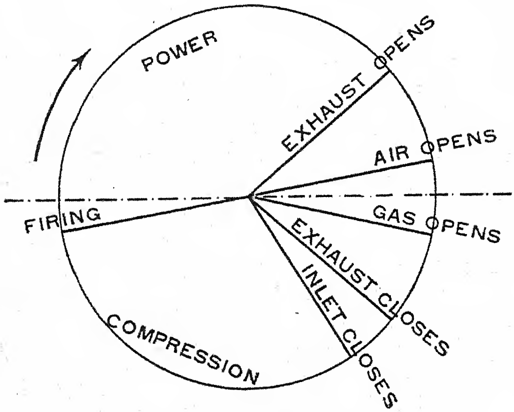


FIG. 205.

The above drawbacks do not, however, apply to the two-stroke compression-ignition oil engine, in which air only is used for scavenging, and a large number of two-stroke oil engines for large and small horsepowers are in use for speeds up to 500 r.p.m.

A two-stroke engine with improved scavenging arrangements is described later (Figs. 219 and 220).

Fig. 205 refers to the crank positions of a large Körting two-stroke cycle, double-acting gas engine.

The mixture of gas and air in the cylinder is fired, and the piston is urged forward.

When the piston is near the end of its working stroke,

the crank being in the position shown, the exhaust valve is opened. The pressure in the cylinder is rapidly reduced to atmospheric pressure. The air valve opens just before the crank reaches the outer dead centre, and admits air at about 9 lb. pressure above the atmosphere, supplied by a separate pump. The effect of this scavenging air is to cool the burnt products, and so to minimise the danger of pre-ignition when the new charge is admitted, as well as to give a better burning mixture. When the crank has passed the dead centre, the gas valve opens and admits gas under pressure along with the air. The exhaust valve closes soon after the gas valve has opened, and lastly the gas and air inlet valves both close. Compression of the mixture by the engine piston now takes place until the firing point is reached in the cylinder when the crank is near the inner dead centre.

The *indicator diagram* from a gas engine may be obtained by an ordinary indicator. An outside spring should preferably be used owing to the high temperature of the gases.

Fig. 206 shows the normal indicator diagram from a four-

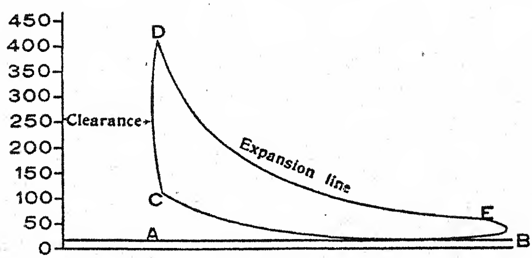


FIG. 206.

stroke cycle engine taken with a strong spring. A scale of pressures and the clearance has been added.

Considering the case of a four-stroke cycle, from A to B the charge is drawn into the cylinder, the pressure being slightly below the atmosphere. From B to C compression takes place. At C the mixture is fired, and the pressure rises to D. From D to E expansion of the gas takes place, the exhaust valve opening at E just before the end of the

stroke. EA represents the exhaust which takes place at a pressure slightly above the atmospheric pressure.

The suction line AB and exhaust pressure line appear as one line coinciding with the atmospheric pressure line when taken with the usual strong spring in the indicator.

Fig. 207 shows the suction and exhaust pressure lines as taken by an indicator having a weak spring. The mean

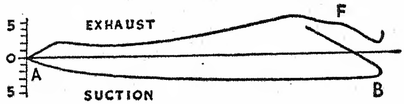


FIG. 207.

pressure on the piston is obtained, as in the steam-engine diagram (see p. 52), by dividing the indicator diagram into ten equal parts or by using a planimeter.

The indicated horsepower is obtained from the formula $\frac{plan}{33{,}000}$, where p is the mean pressure in pounds per square inch on the piston.

l is the length of the stroke in *feet*.

a is the area of the piston in *square inches*.

n is the number of *explosions* per minute.

Example.—The mean pressure on the piston is 80 lb. per sq. in.; stroke, 24 in.; diameter, 1 ft.; 70 explosions per minute. Find the indicated horsepower.

$$\text{I.H.P.}=\frac{80\times2\times12\times12\times22\times70}{33{,}000\times28}=38{\cdot}4$$

The units used should be carefully noted, namely, total pounds pressure multiplied by feet per minute during which the pressure acts and divided by 33,000. Note $\frac{22}{28}=\frac{\pi}{4}$.

In a steam engine, the indicated horsepower and the power actually obtained from the crankshaft, called the brake horsepower, are not very different—the brake horsepower averaging over 90 per cent. of the indicated power.

In an internal combustion engine, however, the brake horsepower, or the power delivered at the crankshaft, is

considerably less than the indicated power, the brake power averaging about 80 per cent. or less of the indicated power. The greater difference is due to the loss by friction during the non-working strokes of the internal combustion engine and the work done in overcoming the suction effect required for the purpose of drawing in the charge of gas and air.

It is therefore more usual to speak of the brake horsepower of a gas engine than the indicated horsepower.

Valves and Valve Gears.—With few exceptions, the inlet and exhaust of internal combustion engines are controlled by poppet valves. These valves are held to their seating by strong springs, and as the valves usually open inwards, the pressure in the cylinder helps to keep them closed. The valves are lifted from their seats and the ports opened either by cams having projecting portions designed to give the period of opening required or by eccentrics operating through link-work. Of these two methods the cam gear is the more commonly used, but in either case it is necessary that the valve gear shaft of an engine should rotate but once from beginning to end of a complete cycle however many strokes may be involved in the completion of that cycle. This is necessary to secure a continuous regulation of the valve gear as required. For this purpose the cams or eccentrics of four-stroke engines are mounted on shafts driven by gearing at half the speed of the crankshaft. The curves used for the acting faces of cams depend on the speed of the engine and the rapidity of valve opening desired. In high-speed engines the inertia of the valve is important, and the cam must be formed having regard to this fact, otherwise excessive shock and noise may result. In gas engines running at low or moderate speeds a usual form of cam is made up of two inclined straight faces joined by a circular arc. Cams of this type have been drawn in the solution of the following example.

1. *Question.*—In a four-stroke cycle gas engine the valves are operated by cams on a shaft which rotates at half the speed of the crankshaft, that is the camshaft rotates through 180° while the crankshaft makes a complete revolution.

Draw an end view of a camshaft with the necessary cams in their correct positions to open and close the valves as shown in Fig. 201 (p. 275). Show the direction of rotation by an arrow.

Solution.—The table below gives the position of the cams relatively to that of the crank. The angles are all measured from the zero position taken at the beginning of the cycle, i.e. the inner dead centre of the crank.

	Crank position	Camshaft position
Inlet valve opens . .	0° . . .	0°
Inlet valve closes . .	198° . . .	99°
Exhaust opens . . .	516° . . .	258°
Exhaust valve closes .	720° . . .	360°

In Fig. 208 (*a*), the two cams and the crank are all in

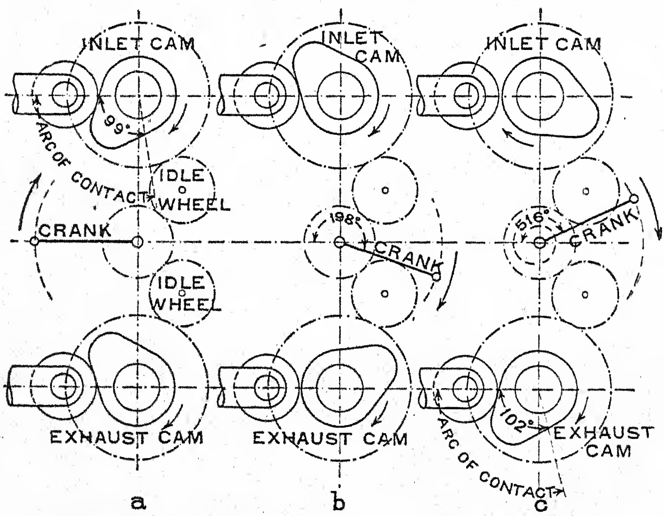


DIAGRAM OF RELATIONS OF CRANK AND CAM POSITIONS.

FIG. 208.

position for the beginning of the cycle, the inlet valve is just beginning to lift, and the exhaust valve is just closed. Fig. 208 (*b*) shows the crank after it has revolved through

198°. The inlet cam has just allowed the inlet valve to close, and the suction period at the end of the first stroke is completed. Both the inlet and exhaust now remained closed during the greater portion of the following two strokes until the position of the crank shown in Fig. 208 (*c*), when the exhaust begins to open. The end of the exhaust stroke is shown in Fig. 208 (*a*), since it coincides with the beginning of admission. In the diagrams the two cams have been shown on separate shafts to avoid confusion, but it is usual to mount the cams side by side on one shaft.

The Gas Engine.—Fig. 209 shows a section of a medium size Hornsby-Stockport engine, made by Messrs. Richard Hornsby & Sons, Ltd., Grantham. In this example, the cylinder liner is cast separate from the jacket, and the combustion chamber is bolted to the end of the main casting. The gas admission valve G is opened when the knife-edge A rises and moves the rod B upwards, and so causes a bell-crank lever, which is partly shown in the figure, to move the gas valve to the left. If the speed rises above the normal speed, the governor moves the knife-edge forward, so that it just misses the rod B, and the gas valve is not opened.

The rod D is moved upwards at the same time as the knife-edge A, and lifts one end of the lever F, the other end depressing the combined air and gas inlet valve H. The time of opening and the duration of the opening of the valves are controlled by cams on the two to one shaft. All the valves are closed by springs.

The Oil Engine.—The oil engine is an internal combustion engine which takes oil fuel into its working cylinder in the liquid state. The oil is sprayed into the cylinder with or without the assistance of an independent supply of compressed air through an arrangement of fine perforations known as an atomiser. Although it is similar in action to the gas engine, certain modifications are necessary in order to ensure the complete burning of the fuel, which, unlike a gas, will not diffuse naturally through the air in which it is to burn. It is essential that combustion should be complete, not only because it has a very obvious effect on the

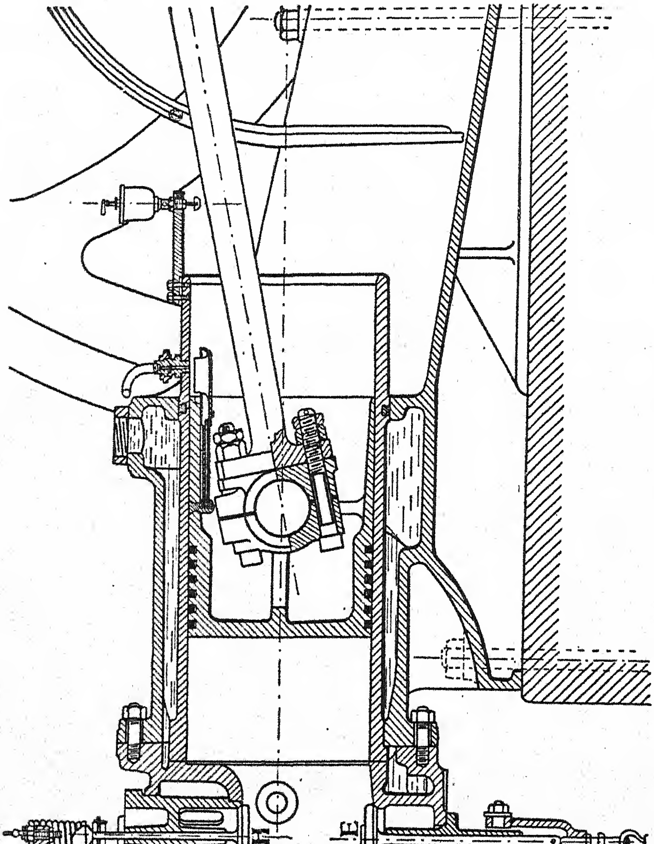


FIG. 209.

efficiency, but also because any unburnt residue is likely to form a deposit on the cylinder walls and thus interfere with the running of the engine. The weight of fuel used per cubic foot of cylinder volume varies from 12 to 30 lb. per hour according to the design of the engine, hence it is clear that even a small percentage of unburnt residue may cause a troublesome deposit in a short time, involving frequent stoppages for cleaning.

In practically all the oil engines now manufactured the fuel used is so-called ' crude ' oil, which actually is the residual portion of the crude oil after the lighter oils such as petrol have been extracted. In these engines air only is admitted during the induction stroke and is compressed to a high pressure and correspondingly high temperature, the final temperature being sufficient to ignite the fuel as it is sprayed into the cylinder at or near the end of the compression stroke. For this reason such engines are now known as ' compression-ignition ' engines.

In the original Diesel engine the fuel was sprayed into the cylinder by means of compressed air at a high pressure (about 1200 lb. per sq. in.). This involves the use of an air compressor, which is costly and takes power for compressing the air, but in the early days of this type there was no satisfactory way of ensuring satisfactory combustion and low fuel consumption. Many large stationary and marine engines still employ this principle.

The search for simpler and equally economical methods of injecting the fuel led to the development of engines with airless injection (sometimes wrongly called ' solid injection '), and in modern engines there is little to choose between the two methods so far as fuel consumption is concerned. In this type pressures varying from 2000 to 10,000 lb. per sq. in. are used in the fuel pump, depending mainly upon the size of the engine.

For satisfactory and rapid combustion of the fuel two conditions must be satisfied :

1. The fuel must be *atomised*, i.e. it must be split up into extremely fine particles.

2. The atomised fuel must be well mixed with air, so that oxygen shall have access to each particle of fuel.

In the Diesel engine the sudden expansion of the compressed air which is mixed with the fuel on entry to the cylinder splits up the fuel and effects atomisation.

With airless injection the high velocity of exit of the fuel from the fuel nozzle due to high pressure effects the necessary atomisation.

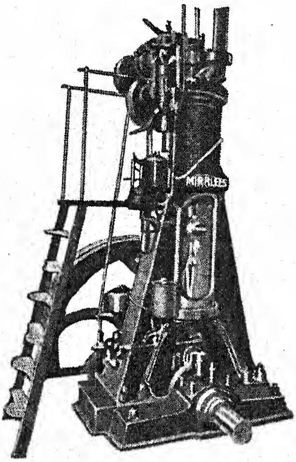


FIG. 210.

In the case of large slow-speed engines a very high fuel pressure is employed in order to give the atomised fuel sufficient velocity to penetrate throughout the combustion chamber and so mix with the air. Unfortunately, the finer the atomisation is the less will be the penetration, and for this reason a large excess of air is necessary to ensure complete combustion.

With high-speed compression-ignition engines the *air* itself is given a turbulent or whirling motion usually by a special construction of the cylinder head, and this has the advantage that as the speed increases the increased turbulence speeds up the combustion.

Fig. 210 shows the general appearance of the Mirrlees Diesel oil engine. One working cylinder is shown, and a small two-stage air compressor driven by a separate crank.

The engine illustrated was the *first* cold starting Diesel engine built in Great Britain (in 1897) and *third* in the world.

It is worked on the Otto four-stroke cycle, as follows :

1*st Stroke*.—Pure air is drawn into the cylinder.

2*nd Stroke*.—The air is compressed to nearly 500 lb. per sq. in.

3rd Stroke.—A fine spray of oil is injected into the highly heated compressed air during a short portion of this stroke by means of air compressed to a pressure of about 650 lb. per sq. in. This compressed air is provided by the small two-stage air compressor attached to the engine. The air and oil burn at constant pressure, and the products of combustion expand as shown by the indicator diagram.

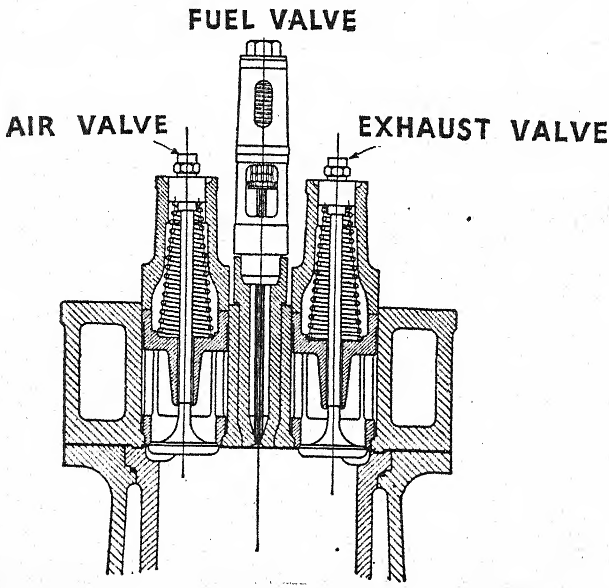


FIG. 211.

4th Stroke.—The products of combustion are expelled from the cylinder.

No hot tube or electrical ignition is required, and there is no danger of pre-ignition. A cheap oil may be used. The thermal efficiency of the engine is very high—one brake horsepower for one hour may be obtained for less than half a pound of oil.

Fig. 211 shows a section of one of the working cylinders. Poppet valves are used for the air and exhaust, and are

depressed by levers worked by cams from the two to one shaft.

The fuel valve is lifted at the right instant, and a spray of oil injected into the cylinder.

The engine is started by compressed air, which is admitted by a fourth valve not shown in the figure. Full load may be taken up in about one or two minutes from starting. The clearance is small in order to secure the high compression of the air.

The Diesel oil engine is built on the two-stroke cycle also. In this form the valve gear may be simplified by the omission of the air inlet valve and exhaust valve, and the weight per horsepower may be a little reduced, although the weight of a two-stroke cycle engine is considerably more than half that of a four-stroke cycle engine of equal power. A pump for the scavenging air is necessary for the two-cycle engine.

Fig. 212 shows an indicator diagram from a four-cycle Diesel engine. It will be seen that the compression reaches about 500 lb. per sq. in., after which the pressure remains nearly constant from *c* to *d* in the firing stroke. From *d* expansion takes place until the exhaust port is opened at *e*. The dotted line shows how the engine is governed by reducing the length of the oil injection period for lighter loads.

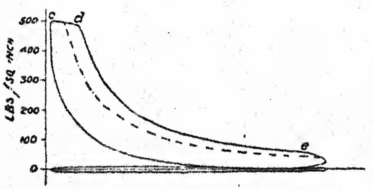


FIG. 212.

In using certain fuels, such as tar oil, which have a very high temperature of ignition, it has been found that the compression must be inconveniently high if that alone be relied upon to cause ignition. Petroleum can be used with compression below 500 lb. per sq. in., but tar oil requires 600 lb. to ignite it on starting the engine. To avoid this a pilot injection or ignition charge of petroleum is employed, not only for starting but also for continuous running.

Figure 213 shows how this is done by the makers of the Mirrlees Diesel engine. Two fuel pumps are used, one of which delivers a charge of petroleum (from 5 to

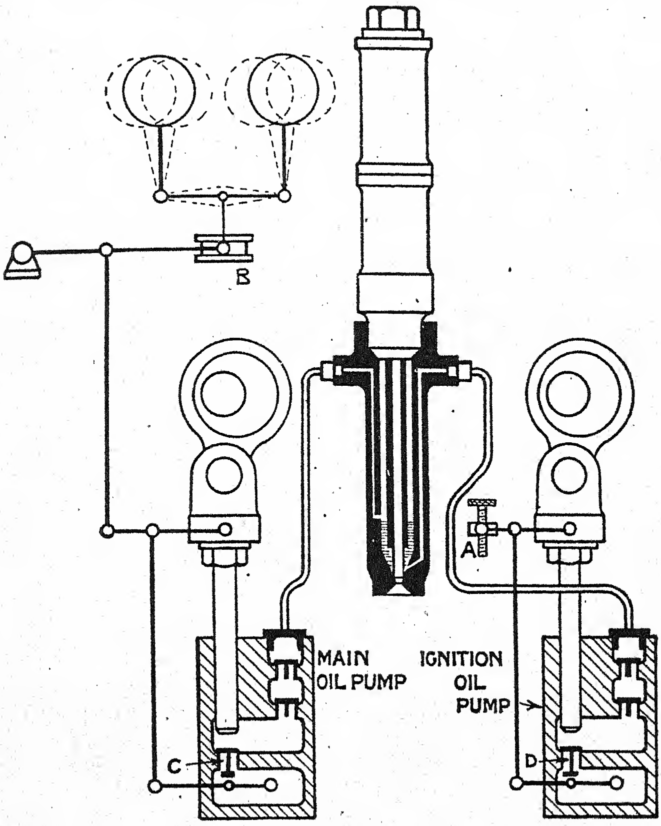


FIG. 213.

10 per cent. of the full-load fuel consumption) near to the bottom of the fuel valve, the other pump delivers the main charge of tar oil above the ignition charge. Thus, when the fuel valve is opened the petroleum first enters the cylinder, burns and raises the temperature to a point high

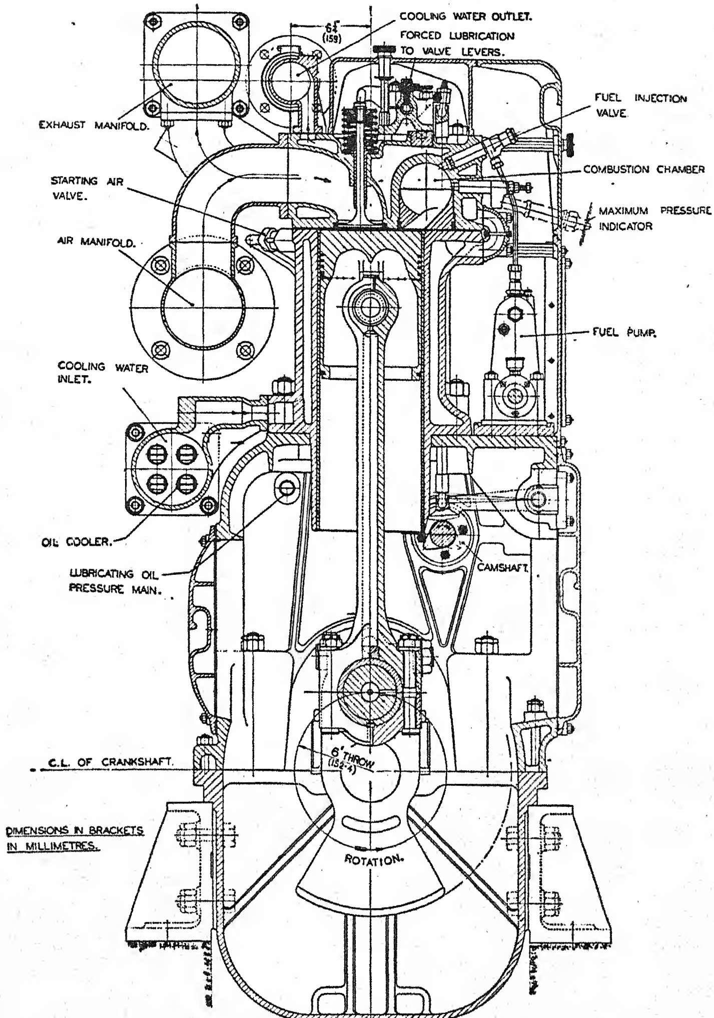
enough to ignite the tar oil which immediately follows. The amount of ignition oil remains almost constant at all loads, being adjusted by the screw A, which determines the closure of the suction valve on the ignition-oil pump. The main injection of tar oil is regulated by the governor through the device shown, whereby the suction valve of the main fuel pump is held open for a longer or shorter time, so that more or less oil is returned to the reservoir instead of being pumped into the atomiser.

Figs. 214 and 215 show transverse and longitudinal sections of a modern high-speed compression-ignition engine, manufactured by Mirrlees Bickerton and Day, Ltd. The engine runs up to 900 r.p.m. and develops 60 B.H.P. per cylinder at this speed.

This engine is of 8 in. diameter by 12 in. stroke operating with a mean pressure of 87 lb. per sq. in. and a piston speed of 1800 ft. per minute at its highest rating. The engine is fitted with a Ricardo Comet type combustion chamber, which gives good combustion and clean exhaust at these high speeds of operation, and there is no fouling of the exhaust valve due to unburnt carbon, and since a single-hole Pintle type nozzle is used in this combustion chamber there is no tendency for the nozzle holes to become choked, as in the case of nozzles having a number of holes of small diameter.

On the upstroke of the piston air is compressed into the spherical chamber shown, and thus is given a vigorous whirling motion, which atomises the fuel very completely as it issues from the nozzle. A portion of the spherical chamber is left unjacketed as shown and the heat of this assists in rapid vaporisation of the fuel. Partial combustion takes place in the chamber and is completed early in the down stroke of the piston as the gases issue from the chamber. The higher the speed of the engine the higher is the speed of whirling of the air and the more rapidly atomisation and mixing takes place.

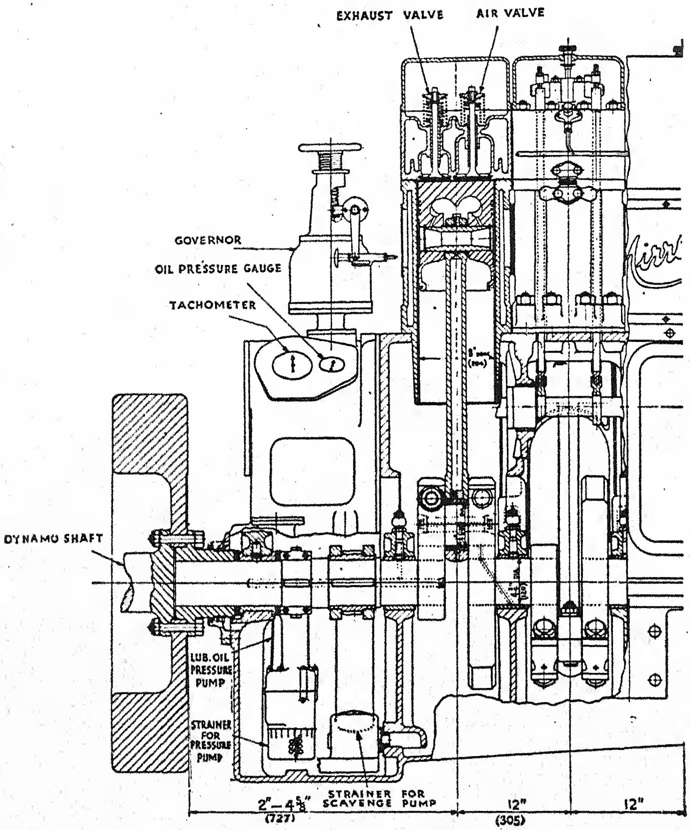
The cylinder head is arranged with a large air valve and smaller exhaust valve, which gives the maximum volumetric efficiency.



8" × 12" "MIRRLEES - RICARDO" HIGH SPEED ENGINE. CROSS SECTIONAL ARRANGEMENT.

FIG. 214.

The engine has a monobloc crankcase on which separate cylinders are bolted. This construction facilitates accurate manufacture and alignment of the cylinder bores with the crankshaft, and also results in easy replacement of any cylinder should one be accidentally broken or should one



5-CYL. 8" × 12" "MIRRLEES-RICARDO" HIGH SPEED ENGINE ("COMET-HEAD" TYPE) LONGITUDINAL SECTION OF ENGINE.

FIG. 215.

be cracked by an attendant carelessly omitting to drain the engine during frosty weather. These separate cylinders and cylinder heads permit of unit construction and manufacture so that engines of from two to eight cylinders can be built up from the same parts. The cylinders are fitted with hardened dry liners so that there is no trouble from water joints at either end of the cylinder and the best possible wearing material can be chosen for this liner.

The crankshaft has its pins and journals of Nitrided steel and is supplied with forced lubrication from a Mirrlees pump.

The engine bed operates on the dry sump system, a scavenge pump being employed for pumping the oil from the crank chambers into the end section of the bed nearest the flywheel, which performs the functions of an oil tank. Both the scavenge pump and the pressure pump are mounted in this end section of the bed with filters on the suction sides of the pump. The lubricating oil is pumped by the pressure pump through a pressure regulating valve which bye-passes excess oil into the bed, and the oil to the bearings flows through a duplicate filter and oil cooler.

Air starting can be effected to all cylinders and is controlled by an air distributor valve operated from the forward end of the camshaft.

The valve gear is operated from the camshaft through push rods and the monobloc type fuel pumps are driven from the camshaft through gearing.

The Hot-bulb Engine.—A somewhat lower compression can be used in an engine using heavy oils if a portion of the cylinder end is left unjacketed. Such engines are known as 'hot-bulb engines,' and it is necessary to heat the bulb by means of a blowlamp before the engine is started. A number of these engines (formerly called 'semi-Diesel' engines) are in use, but they are now rarely manufactured.

Figure 216 shows a vertical section through an engine of this kind as made by Messrs. W. H. Allen, Son and Co., Ltd., of Bedford.

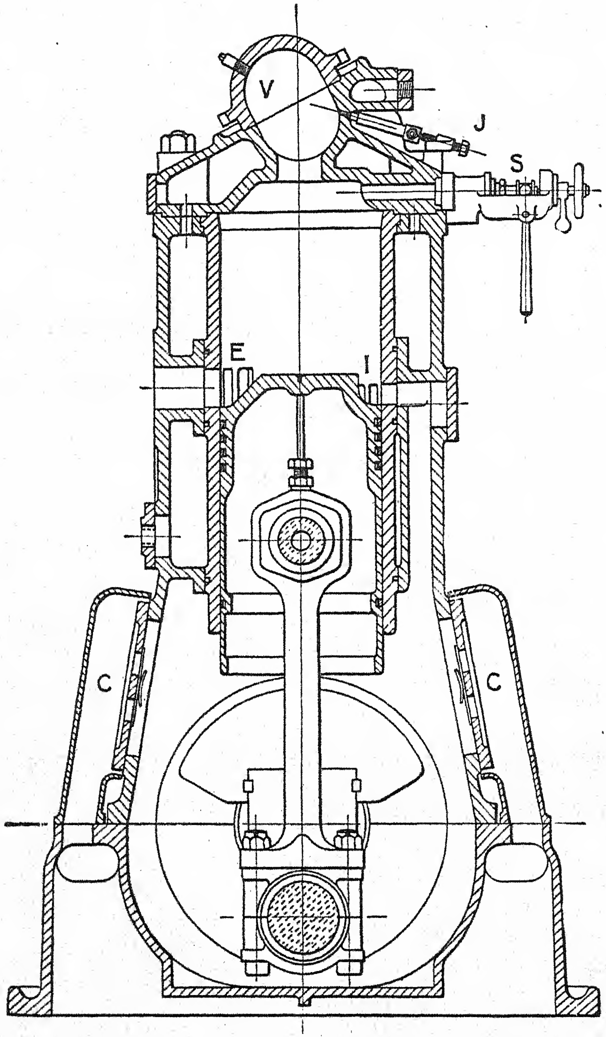


FIG. 216.

During the downstroke of the piston air is slightly compressed in the crankcase, as already explained in Figs. 202 to 204, until the ports marked I are uncovered by the piston nearly at the end of its travel. The air from the crankcase, under slight pressure, then rushes into the cylinder and helps to expel the burnt gases through the previously opened exhaust ports E. This action is assisted by the peculiar shape of the piston face, which is designed to deflect the air upwards and keep it from taking a direct path to the exhaust ports. As the piston returns on its upward stroke the air inlet ports are first closed and shortly afterwards the exhaust ports are closed. Compression of air alone follows until, nearly at the end of the upstroke a charge of oil is sprayed into the heated bulb V at the head of the cylinder and ignition takes place, due to the combined action of the heated surface of the bulb and the compression of the gas. The pressure rises and the expansion stroke follows until the exhaust ports are once more uncovered by the piston. Inlet valves C admit air to the crankcase during the upstroke, and the cycle is repeated. In this engine compressed air is not used to blow the oil into the cylinder, but the oil itself is pumped under high pressure through a special jet or atomiser. The bulb is heated externally for the purpose of starting, but once it is heated no further application of heat is necessary. Fig. 217 shows an indicator diagram taken from an engine of this kind at full load. The form of the end of the diagram from *a* to *b* is characteristic of the two-stroke cycle, and shows the exhaust and charging period explained also in Figs. 202 to 204.

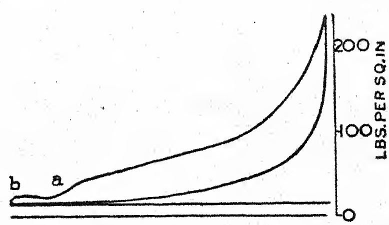


FIG. 217.

Difficulty is sometimes found when working with hot-bulb engines on very variable loads: at light loads the bulb

tends to cool and ignition becomes uncertain ; at heavy loads the bulb may become overheated and the ignition takes place too suddenly. A device introduced by Messrs. Blackstone & Co., Ltd., of Stamford, to give steady firing

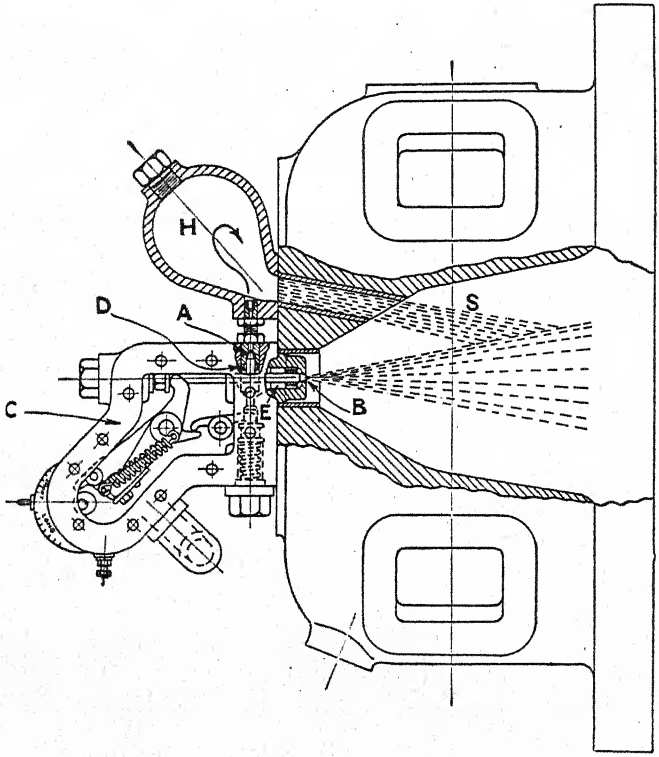


FIG. 218.

at all loads is shown in Fig. 218. Two fuel valves are used, A and B. The supplementary valve A opens slightly before the main valve and admits a definite quantity of oil to the bulb H. This is fired by the heat of the bulb, and a stream of flame S is projected forward to meet the main

oil supply from the valve B. The main oil supply is greater or less according to the load, but a constant and sufficient amount is always sent to the bulb so that its heat is maintained. By this arrangement the heavier residual oils are used without difficulty, not only at full but at light loads.

The oil inlet valve gear is enclosed in an airtight casing C, which is supplied with air at a pressure of about 400 lb. per sq. in. Thus when the valves are raised from their seats the compressed air flows towards the cylinder and hot bulb. But the passages leading past the inlet valves contain the fuel oil which has already been placed there by the oil pump in readiness, and the high-pressure air in escaping towards the cylinder carries with it the charge of oil. The fuel oil from the pump reaches the main oil passage by way of a number of fine holes in the block D, which forms a guide for the valve A. A definite quantity of oil is sent to the hot bulb each time the inlet valve A is lifted. The remainder of the oil, which is varied to suit the load, goes to the cylinder by the main inlet B, and is ignited by the flame coming from the oil already burning in the bulb. Hand adjustments are provided by which the time of opening of the valves may be changed to suit particular conditions.

A small two-stage air compressor mounted at the side of the bed provides the air for oil injection and for starting.

Figs. 219 and 220 show a vertical section and sectional elevation of a modern two-stroke cycle engine as made by Messrs. Petters, Ltd., of Yeovil, known as the 'Super-scavenge' type. The scavenging air is furnished by a rotary blower, gear driven from the engine crankshaft. This delivers air at about $1\frac{1}{2}$–$1\frac{3}{4}$ lb. per sq. in. to an air trunk and thence to a ring of ports located at the bottom of the piston travel. The cylinder head is furnished with two exhaust valves operated by push rods from a camshaft driven at engine speed, the train of gears including the blower drive. These two shafts are interconnected by suitable idler wheels. The camshaft operates also the

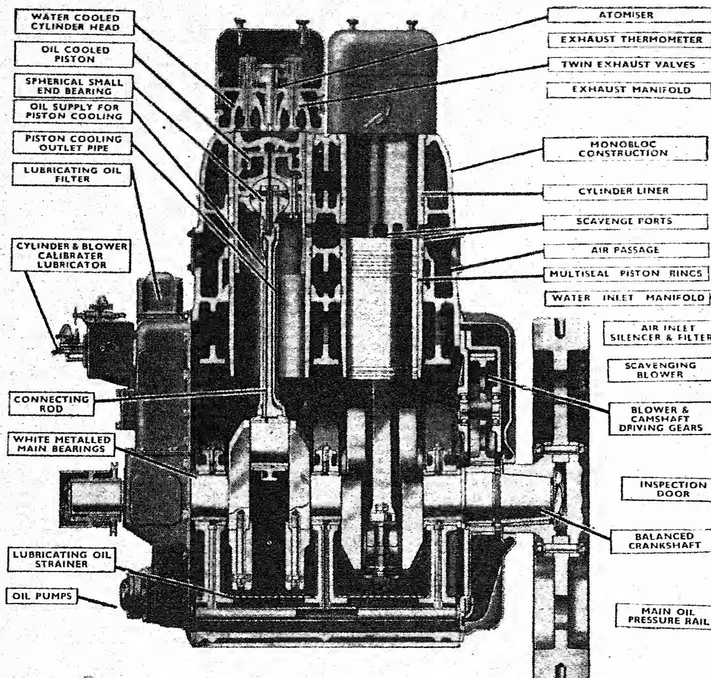


FIG. 219.

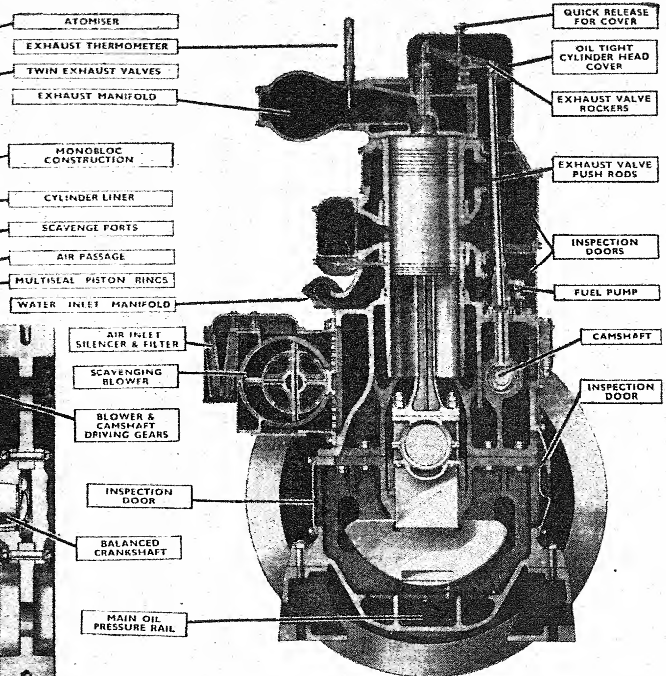


FIG. 220.

C.A.V.-Bosch fuel pumps, one for each cylinder and the necessary air-starting valves.

The engine lubrication operates on the dry sump system, two oil pumps being provided. The pressure pump delivers oil from a tank to the main bearings, connecting rod large end bearings, and pistons. The latter are cooled by the circulation of a generous supply of lubricating oil. The camshaft bearing and all other working parts are supplied with pressure feeds. The oil returning to the sump of the engine is picked up by the second pump, and returned through an oil cooler to the supply tank. The cooling water for the engine water jackets is passed through the oil cooler on its way to the jackets.

The jackets and cylinder housing are of monobloc construction with loose 'wet' liners. The construction of this housing together with the bedplate form a very rigid construction. Included in the cylinder housing are a sensitive governor driven from the engine camshaft, a lubricator for controlling the oil supply to the cylinder liners and an oil filter of the 'Streamline' type. The latter is supplied with a part only of the oil delivered by the pressure oil pump. The clean oil is delivered to the lubricator, any surplus being returned to the sump. The whole of the oil is continuously being purified in this manner.

The cycle of operations is as follows. The preceding charge having been fired, the piston descends towards the latter part of its stroke, and before the air ports at the bottom of the cylinder are reached, the exhaust valves are opened. When the pressure in the cylinder has fallen to nearly atmospheric, the air ports open. The exhaust valves are arranged to close slightly before the air ports. By this means the cylinder is cleared of the burned gases and is completely filled with cool clean air. Thus when fuel injection takes place towards the end of the compression stroke, *complete* combustion is obtained, resulting in an invisible exhaust at all conditions of speed and load.

The engine illustrated has a bore of 8·5 in. by 13 in.

stroke, and at a speed of 500 r.p.m. develops 62·5 B.H.P. per cylinder. It is manufactured in units of two to six cylinders, giving powers from 125 to 375 B.H.P. It is available as an industrial unit for general power and electrical generating purposes and also as a marine and traction unit.

The guaranteed fuel consumption at the rating above mentioned is only 0·39 lb. per B.H.P. per hour with a lubricating oil consumption of only 0·003 pint per B.H.P.

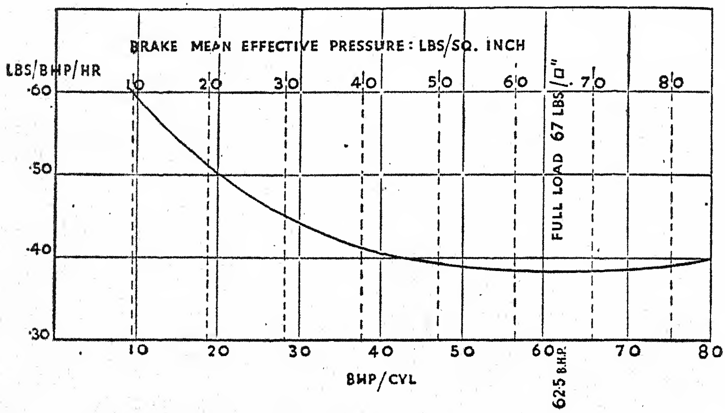


FIG. 221.

per hour. At the rated load the B.M.E.P. is 67·1 lb. per sq. in. The engine is capable of high overload due to the very complete combustion obtained with invisible exhaust. The exhaust temperature is very moderate being about 700°–750° F. at full load.

Fig. 221 shows the variation of specific fuel consumption with load. It will be seen from this that the specific fuel consumption (lb. per B.H.P. hour) is practically constant for all loads from 40 B.H.P. to 80 B.H.P. per cylinder. Since the mechanical efficiency decreases with decrease of load this shows that the fuel per hour per indicated horsepower becomes *less* as the load decreases from 80 B.H.P.

to 40 B.H.P. This is due to the fact that the combustion temperature decreases as the quantity of fuel injected per working stroke decreases, thus reducing the heat losses to jacket and exhaust.

Fig. 222 shows an indicator diagram taken at full load and Fig. 223 is a diagram taken on a time base with a

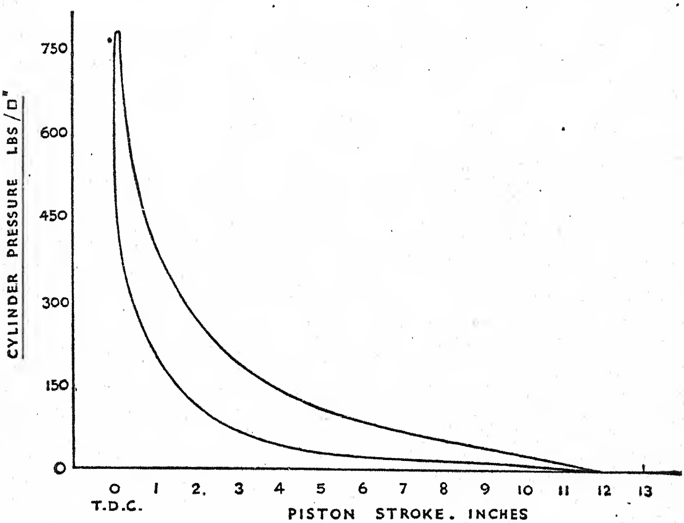


Fig. 222.

'Farnboro' indicator, showing how the pressure rises during injection of the fuel.

The mechanical efficiency with a brake mean effective pressure of 67 lb. per sq. in. is 83 per cent.

Amongst the novel features of the design is a spherical small end bearing, the oil-cooled pistons allowing of long periods of overload without overheating. Cylinder distortion is avoided by the adoption of the unidirectional system of scavenging, with the exhaust valves in the cylinder heads.

A short stiff crankshaft ensures relative freedom from

torsional vibrations. The engine starts instantly from cold. The piston speed is moderate, being only 1082 ft. per minute at the normal speed of 500 r.p.m.

Amongst other recent products of Messrs. Petters, Ltd., may be mentioned the 'Harmonic' induction engine. The layout of this engine is somewhat similar to the 'Super-scavenge' engine, but in this case the scavenging air is provided automatically and without the aid of a blower, the energy of the exhaust gas being utilised to create an induction effect on the atmospheric air. The bore of this engine is $4\frac{1}{2}$ in. with a stroke of $6\frac{1}{2}$ in. and 16 B.H.P. is developed at 1000 r.p.m. The fuel consumption is below 0·4 pint per B.H.P. per hour, the lubricating oil consumption being less than 1 per cent. of the fuel consumption. This engine is of course of the compression-ignition type.

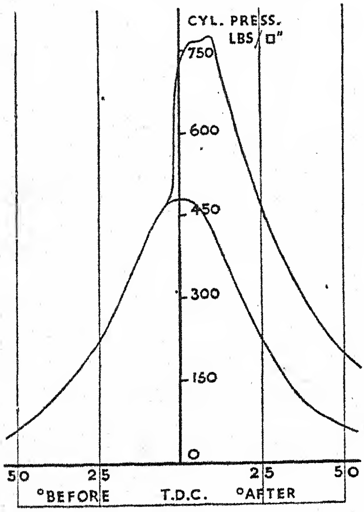


FIG. 223.

The thermal efficiency of the internal combustion engine depends very largely on the amount of compression given to the charge before ignition. The higher the compression, the greater the possible efficiency. Fig. 223A shows how the efficiency increases as the compression is increased from 2 to 10, or, in other words, the volume is reduced by compression to $\frac{1}{2}$ or $\frac{1}{10}$ before ignition.

The efficiencies shown by the curve are for an engine in which no heat is lost to the cylinder walls during compression, ignition, or expansion. Since in actual practice heat is lost in each of these ways the actual thermal effi-

ciencies obtained are appreciably lower, but the general shape of the curve still shows the way in which the efficiency increases with the compression ratio. In modern gas and petrol engines the compression ratio varies from 5 to 8, but in the case of the compression-ignition engine a com-

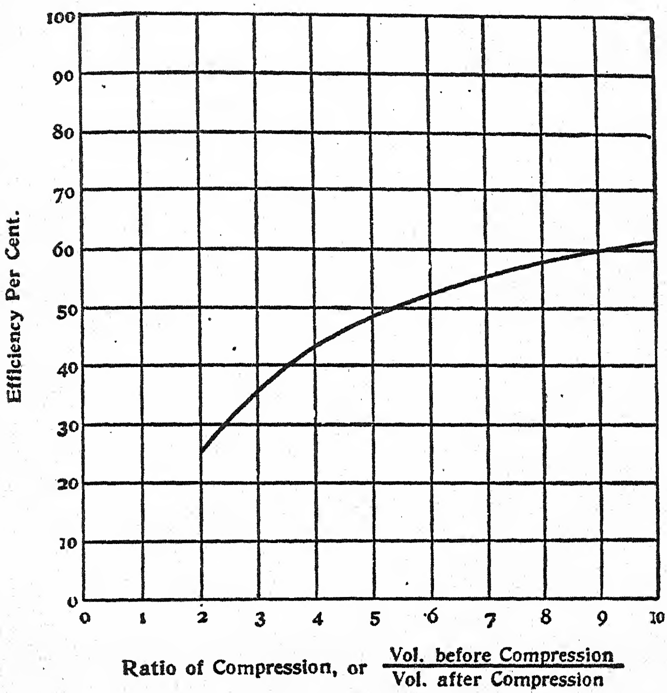


FIG. 223A.

pression ratio of from 10 to 14 is necessary in order to raise the air to ignition temperature. This is one reason for the superior efficiency of this type of engine.

The effect of compressing the mixture to a higher pressure before ignition is shown by the dotted area in Fig. 224, which represents the added work from the higher compression, the clearance being reduced from *a* to *b*.

The advantages of high compression may be briefly stated as follows :

1. A greater mean pressure on the piston every working stroke.

2. A smaller size of cylinder for the same horsepower.

3. A weaker mixture may be used and fired with more certainty.

4. Less burnt gases will be left in the cylinder at the end of the exhaust stroke.

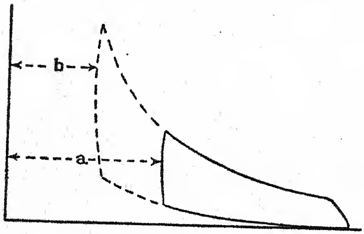


FIG. 224.

5. The engine will require less fuel for the same power.

The great objection to compressing an explosive mixture beyond a certain amount is that the mixture may explode before the end of the compression stroke, thus causing a loss of power and a heavy stress on the engine. The cause of the pre-ignition is due to the high temperature developed in a gas when it is compressed. Some gaseous mixtures fire at a lower temperature than others. A gas containing chiefly carbon monoxide (CO), such as producer gas, may be compressed to a higher pressure without pre-ignition than a gas containing much hydrogen, like ordinary coal gas.

Another objection to a high compression in a gas or petrol engine is that the maximum pressure is increased (see Fig. 224). Since the engine must be designed for the maximum pressure the weight and cost are increased.

Reference to Fig. 223A shows that the gain in thermal efficiency due to increased compression is less for high ratios than for smaller ratios.

The indicator diagram of an internal combustion engine may be used to find the temperature of the gas at any part of the cycle, if the temperature at any one point is known.

Take an indicator diagram (Fig. 225) having an atmospheric line CD, and draw OX, the zero line, to the same scale as the diagram, 14·7 lb. below the atmospheric line.

If AB is the length of the diagram and the clearance is 20 per cent., make OA 20 per cent. of AB, and draw OE to represent the clearance line.

From the properties of a gas we know that PV=RT, where P, V, and T represent the pressure, volume, and *absolute* temperature of a gas, and R is a constant.

Assume that the temperature at D at the commencement

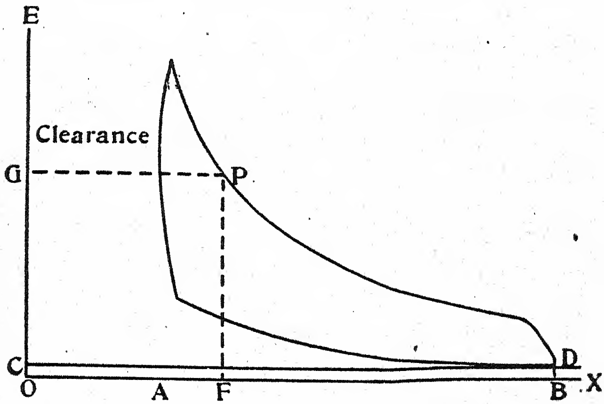


FIG. 225.

of the compression stroke is known. If the mixture did not receive any heat from the hot cylinder, and from the residue of gas left in the cylinder, the temperature would be about 60° F. It receives, however, heat from the cylinder, and its temperature may be assumed to be, say, 200° F.

Measure DB, the pressure at D, and the volume DC. As these quantities are to be compared to similar quantities, any scale of measurement will suffice, as long as the remaining quantities are measured to the same scale. For instance, DB may be measured in pounds, or in inches, or in millimetres. Similarly, DC may be measured in inches or millimetres.

Suppose DB=0·07 in. and DC=2·79 in.; then, substituting in the equation $\frac{PV}{T}=R$, we have

$$\frac{0{\cdot}07\times2{\cdot}79}{200+461}=R, \text{ and } R=0{\cdot}000295$$

To find the temperature at any other point P during expansion.

Measure PF=1·08 in. Measure PG=1·05 in.

As R is the same as before—

$$\frac{1{\cdot}08\times1{\cdot}05}{T}=0{\cdot}000295$$

$$\therefore T=3838^\circ \text{ abs., or } 3377^\circ \text{ F.}$$

Since compression-ignition oil engines are in use for commercial road vehicles, and their use is likely to be extended very greatly in the future, a brief summary of the advantages and disadvantages of the compression-ignition engine for road vehicles is given below.

Advantages.

1. Lower weight of fuel consumed. The fuel consumption per brake horsepower-hour is about two-thirds of that of the petrol engine at full load and is still more favourable at lighter loads.

2. Still lower *volume* of fuel consumed, owing to the higher density of fuel oil. One gallon of fuel oil represents about 20 per cent. more heat than one gallon of petrol.

3. Low cost of fuel per gallon. The addition of the tax on fuel oil has made a material difference to the relative costs, but petrol is still 50 per cent. dearer than fuel oil.

4. Safer fuel. Owing to its higher flash-point, fuel oil does not readily give off inflammable vapour.

5. More complete combustion and consequently less carbon deposit and objectionable exhaust fumes.

6. A much greater range of mean effective pressure possible, so that the engine is more flexible. The smallest practicable mean effective pressure for a petrol engine is never less than 70 per cent. of the maximum. A com-

pression-ignition engine can work between mean effective pressures of 35 and 115 lb. per sq. in.

7. The engine works at *full throttle* all the time and the power is regulated by the amount of fuel injected per working stroke. Thus there are higher volumetric efficiencies and reduced pumping losses at all loads.

8. Better fuel distribution with multi-cylinder engines. A larger induction pipe is possible, since it deals with air only.

Disadvantages.

1. The ratio of maximum pressure to mean pressure is appreciably higher than for petrol engines, so that the weight and cost per horsepower are greater.

2. Owing to the high maximum pressure, the reciprocating parts are heavier than for petrol engines. This increases the inertia forces.

3. Owing to difficulties of injection and atomisation of the fuel, very high speeds of revolution, such as are common in petrol engines, are not at present attainable. Considerable advance has, however, been made in this respect in recent years.

The advantages of oil engines for marine propulsion may be summarised as follows :—

1. The coal consumption per shaft horse-power at full load with steam propulsion is rarely less than 1 lb. and generally exceeds this. If fuel oil is used for steam raising the consumption is about two-thirds of the above. With oil engines the fuel consumption is about 0·4 lb per s.h.p. hour at full load and varies very little from full to half load. Thus for a vessel of 5000 s.h.p. the fuel consumption for a run of 500 hours would be 450 tons as compared with 1200–1400 tons of coal if propulsion is by steam. The saving in bunker space allows for increased cargo capacity.

2. Fuel oil can be stored in the double bottoms of the ship and pumped to the fuel tanks as required.

3. Oil engines take up appreciably more space than steam engines or turbines of the same power, but since boilers are not required there is in most cases a definite saving in total space occupied and also in personnel.

4. There are no 'stand-by' losses, since the engines can be started up from cold and put on to full load in a comparatively short time.

5. Fuel oil can be taken on board while cargo is being unloaded and loaded, so that the ship can be 'turned round' in an appreciably shorter time than with coal fuel.

6. Owing to the system of governing, engines run more steadily in rough seas and a better average speed is maintained in bad weather.

Fuels used in Internal Combustion Engines.—The *gaseous fuels* used in this country are almost all produced from coal, either directly or as a by-product incidental to some process in which coal is used.

Coal gas as used for illumination in towns is very serviceable for power production, especially in the smaller sizes of gas engine ; in larger quantities some cheaper kind of gas such as producer gas is preferred. The calorific value of coal gas varies, but is in the neighbourhood of 500 B.Th.U. per cubic foot of the gas measured at 60° F. and 14·7 lb. per sq. in.

Coke oven gas is very similar to coal gas in quality and calorific value. It is produced in the manufacture of the coke required for metallurgical purposes.

Producer gas is of much lower calorific value than those mentioned above. There are two kinds of gas producer in common use—the 'Suction' producer and the 'Pressure' producer. In both the same principle is involved, namely the incomplete burning of carbon to produce carbon monoxide, which is capable of yielding heat when fully burnt to carbon dioxide. The heat evolved in the producer is partly absorbed in the decomposition of water vapour into hydrogen and oxygen ; the gas thus made being mixed with the carbon monoxide augments its heat value. There is a large proportion of inert nitrogen in producer gas, which is derived from the air used to burn the carbon. Hence the calorific value of producer gas is low, varying from 130 to 160 B.Th.U. per cubic foot at 60° F. and 14·7 lb. per sq. in. pressure according to the type of producer and the fuel used. Coal or coke is the most usual fuel employed, but many kinds of waste material containing a fairly large proportion of carbon have been used with good results. The *Suction producer* depends upon the suction of the engine to draw the necessary amount of air through the producer, and the gas is made as required. The *Pressure producer* is supplied with air under slight pressure, and

lacks the self-regulating features of the suction producer, but this is not in every case a disadvantage.

Blast furnace gas consists chiefly of carbon monoxide and nitrogen. Its heat value is low, being about 100 B.Th.U. per cubic foot at 60° F. and 14·7 lb. per sq. in., but it readily burns in a suitable gas engine.

Liquid fuels are mainly derived from the mineral oils found in different parts of the world. They are distilled from the crude oil, and vary from the light volatile liquid known as petrol to the heavy residual oil from which all the more volatile parts have been removed. The net calorific values range from 18,500 B.Th.U. per lb. for petrol down to about 17,400 B.Th.U. per lb. for a heavy fuel oil.

There are also liquid fuels derived from coal which are coming into more general use on account of the high cost of petroleum products. These fuels are tar oil and benzol. *Tar oil,* net calorific value about 16,500 B.Th.U., is derived from coal tar, and is a heavy oil used in engines of the Diesel and High Compression Solid injection types. Owing to the high ignition point of tar oil special arrangements are necessary for its use. *Benzol* is used as a substitute for petrol. It may be used in the petrol engine without alteration, but as it does not ignite so readily as petrol a higher compression may be used, with consequent increase of the efficiency. The net calorific value of benzol is about 17,000 B.Th.U. per lb.

Alcohol is used as a fuel in some countries, although not as yet to any great extent in this country. It is used in engines similar to petrol engines, with a rather higher compression than is used with petrol. It is obtained from vegetables such as potatoes, beet, etc. The calorific value of commercial alcohol is about 10,000 B.Th.U. per lb., but depends on the percentage of water in solution.

Petrol Engines.—The cycle of operation of a petrol engine is similar to that of a gas engine, and a petrol engine can be run on either coal gas or producer gas. The chief advantage of petrol as a fuel for engines of motor vehicles is that a considerable number of heat units can be stored in a com-

paratively small space. The calorific value of a pound of average petrol is 18,500 B.Th.U. and a cubic foot of petrol (density 0·8) will generate about 920,000 B.Th.U. when completely burned. Now 1 cu. ft. of coal gas at atmospheric pressure has a calorific of about 500 B.Th.U., so that 1 cu. ft. of petrol (about $6\frac{1}{4}$ gallons) is equivalent to 1840 cu. ft. of coal gas. Even if the gas were stored in cylinders at 1800 lb. per sq. in. a total capacity of about 15 cu. ft. would be necessary to be equivalent to 1 cu. ft. of petrol. A calculation on the above basis will show that gas at 10*d.* per therm (100,000 B.Th.U.) is equivalent to petrol at 15*d.* per gallon. Without the fuel tax petrol would cost appreciably less than this, apart from any other advantages. In some heavy motor vehicles gas producers are used, and in this case the cost of the fuel is very much less.

In motor vehicles it is necessary that the weight of and space occupied by the engine should be small for a relatively high maximum horsepower, and to satisfy these conditions the engine must run at very much higher speeds than an ordinary gas engine (up to 5000 r.p.m. in practice). We thus have a number of problems to solve, (1) to vaporise the petrol quickly and effectively, (2) to mix the petrol vapour with the correct quantity of air, (3) to get as large a *weight* of mixture into the cylinder as possible, since the power developed is proportional to this, (4) to complete the combustion rapidly, (5) to exhaust the burnt products at as low a pressure as possible. At 3000 r.p.m. the time taken by one stroke is $\frac{1}{100}$ second, and if combustion is to be completed during 45° of movement of the crank the time from passing the spark to completion of combustion must not exceed $\frac{1}{400}$ second. At one time this was considered to be impossible, as experiments on explosions of petrol vapour in closed cylinders gave a very much slower rate of combustion. The reason for the success obtained in actual engines is the very high degree of 'turbulence' obtained in the cylinder, so that when the combustion is started by the spark the flaming molecules are quickly whirled to all parts

of the mixture and speed up the combustion enormously, High compression also speeds up combustion considerably in addition to increasing the thermal efficiency.

For details of construction of petrol engines and their accessories students should consult text-books on motor-car engineering, and we shall here only deal with the main principles involved.

Fig. 226 shows a simple type of jet carburetter. When the engine is stationary the petrol in the float chamber B and jet J is kept at a level a little below the jet orifice. When the engine is running air is drawn through the choke tube T and the increase of velocity due to the restriction is accompanied by a fall of pressure, so that the difference between the pressure of the air in the float chamber B and that in the choke tube forces petrol out of the jet orifice, and this is atomised by the stream of air and passes with the air into the cylinder.

If the engine runs at a constant speed the sizes of choke tube and jet can be adjusted to give the correct mixture, but with this simple type the mixture strength increases as the speed increases, so that either the mixture is much too weak for slow speeds or it is much too rich for high speeds. All the apparent complications of modern carburetters are designed to avoid excessive changes of mixture strength with speed.

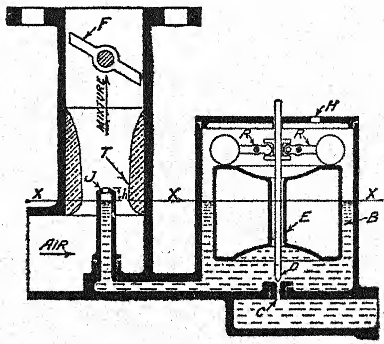


FIG. 226.

As explained previously (see p. 272), in order that the mixture may flow into the cylinder the pressure must be less than the external pressure, and at high speeds the pressure at the end of the induction stroke may be very much below atmospheric, so that the *weight* of

charge induced and consequently the power developed is correspondingly low. This is the reason why the mean effective pressure tends to fall off as the speed increases. The drop of pressure may be reduced (1) by making the valves as large as possible, (2) by the use of overhead valves, giving direct access to the cylinder, (3) by advancing the timing of the valves.

The ratio of the actual weight of charge in the cylinder at the end of the induction stroke to the weight of the same volume at normal atmospheric temperature (15° C.) and pressure (14·7 lb. per sq. in.) is known as the *volumetric efficiency*. This depends upon the *temperature* of the charge as well as its pressure, and this temperature depends upon how much hot gas is left in the clearance volume at the end of the exhaust stroke. With high compression the clearance volume is less, so that the charge will be cooler at the end of the induction stroke. High compression, therefore, will improve the volumetric efficiency as well as the thermal efficiency.

If the mixture is compressed to a higher pressure before entry to the cylinder by means of a *supercharger* the weight of charge taken in is increased accordingly. By this means the mean effective pressure is increased and the falling off of power at high speeds reduced. In the case of aero engines operating at high altitudes, where the density of the air is low, supercharging is necessary to avoid a falling off in power as the height attained increases.

If the time from the passing of the spark to attainment of maximum pressure is the same for all speeds, and maximum pressure is required at or near the beginning of the stroke in every case for best results, then the spark must pass at earlier positions of the crank as the speed increases. Actually the increased turbulence with increased speed reduces the time of ignition, so that the actual advance required does not increase in proportion to the speed.

In petrol engines using high compression ratios the temperature at end of compression is very high, but not high enough in itself to cause spontaneous ignition (known as

'pre-ignition'). If, however, there is a hot spot in the cylinder (sometimes due to carbon deposit) the additional heat due to this will cause pre-ignition (i.e. ignition before the spark passes). In some cases after the spark has passed the heat of the travelling flame causes the pressure to increase sufficiently to ignite the remainder before the flame reaches it and a very rapid explosion occurs, causing a pinking noise due to the sudden blow on the cylinder walls. This is known as 'detonation,' and is quite distinct from pre-ignition. The use of small quantities of 'dope,' such as tetra-ethyl lead, will reduce or eliminate detonation. Blending of different qualities of petrol, or mixing with benzol or alcohol, are very effective also in reducing detonation.

EXAMPLES, WITH SOLUTIONS

1. A four-cycle oil engine of the high-compression airless injection type was found on test to develop 101 B.H.P. per cylinder at 380 r.p.m The cylinder is $14\frac{1}{2}$ in. in diameter, and the stroke is 15 in. Neglect the power required to drive the engine itself, and calculate the mean effective pressure required to develop the given brake horsepower under the conditions mentioned.

Let p=the mean effective pressure corresponding to the brake horsepower in pounds per square inch.

$$\text{Then } 101=\frac{p\times14{\cdot}5\times14{\cdot}5\times0{\cdot}7854\times15\times380}{33{,}000\times12\times2}$$

$$\text{and } p=\frac{101\times33{,}000\times12\times2}{14{\cdot}5\times14{\cdot}5\times0{\cdot}7854\times15\times380}$$

$$=\underline{85\text{ lb. per sq. in.}}$$

That is only a part of the actual mean effective pressure; there is, of course, the pressure required to overcome the engine friction and other losses.

2. In the test referred to in Example 1, the indicated horsepower was found to be 127 per cylinder. Using the dimensions already given, calculate the mean effective pressure required to overcome the resistance of the engine.

The result might be obtained by using the indicated horsepower formula as before, but since the brake mean effective pressure has already been calculated a shorter method is as follows:

Since all other factors remain constant—

$$\therefore \text{ indicated mean effective pressure}=85\times\frac{127}{101}=\underline{107\text{ lb. per sq. in.}}$$

The mean effective pressure required to overcome the resistance of the engine is equal to

$$107-85=\underline{22 \text{ lb. per sq. in.}}$$

That is, of the mean effective pressure of 107 lb. per sq. in. which would be measured on an indicator diagram, 22 lb. per sq. in. pressure is required to drive the engine, and 85 lb. per sq. in. is available for useful work.

3. A sample of producer gas has a calorific value of 151 B.Th.U. per cubic foot at normal temperature and pressure. It requires 1·42 cu. ft. of air under similar conditions for its complete combustion. What is the number of heat units available per cubic foot of the mixture previous to combustion ?

151 B.Th.U. are contained in a mixture of 1 cu. ft. of gas and 1·42 cu. ft. of air, that is, 2·42 cu. ft. contain

$$151 \text{ B.Th.U. or } \frac{151}{2{\cdot}42}=62 \text{ B.Th.U. per cubic foot.}$$

Owing to the presence of a necessary excess of air and to incomplete filling of the cylinder, the number of thermal units in the mixture per cubic foot of cylinder volume would be less than this. But it is found advisable in any case to limit the number of B.Th.Us. of the charge per cubic foot of cylinder volume to 50 in small engines and 40 in large engines. These limits are imposed by cooling difficulties which occur if they are much exceeded.

It should be noted that the number of cubic feet of air to secure complete combustion of a given gaseous fuel varies with the richness of the fuel in combustible material.

4. A Diesel engine is found on trial to develop 300 B.H.P., consuming 126 lb. of oil per hour of a calorific value of 18,000 B.Th.U. per pound, lower value. The indicated horsepower during the trial is 395. Calculate the brake thermal efficiency and the indicated thermal efficiency.

$$\text{The fuel consumption per B.H.P. per hour}=\frac{126}{300}=0{\cdot}42 \text{ lb.}$$

$$\therefore \text{ number of heat units put in per B.H.P. per hour}=0{\cdot}42\times 18{,}000$$

$$\left.\begin{array}{r}\text{No. of B.Th.U. equivalent to one}\\ \text{horsepower for one hour}\end{array}\right\}=\frac{33{,}000\times 60}{778}=2545$$

Thermal efficiency calculated from B.H.P.

$$=\frac{2545\times 100}{0{\cdot}42\times 18{,}000}=33{\cdot}7 \text{ per cent.}$$

$$\text{The fuel consumption per I.H.P. per hour}=\frac{126}{395}=0{\cdot}319$$

Thermal efficiency calculated from I.H.P.

$$=\frac{2545\times 100}{0{\cdot}319\times 18{,}000}=44{\cdot}3 \text{ per cent.}$$

5. The equation to the compression curve of a petrol engine is $pv^{1\cdot28}$=constant. If the bore is 5 in., stroke 6 in., pressures at beginning and end of compression 14 and 110 lb. per sq. in. respectively, calculate the volume of the clearance space.

Let c=volume of clearance space (cubic inches).
Stroke volume=$0\cdot785\times25\times6=117\cdot8$ cu. in.

$\therefore$ volume at beginning of compression $=117\cdot8+c$
pressure ,, ,, ,, $=14$
volume at end of compression $=c$
pressure ,, ,, ,, $=110$

$$\therefore\ 14\times(117\cdot8+c)^{1\cdot28}=110\times c^{1\cdot28}$$

$$\therefore\ \left(\frac{117\cdot8+c}{c}\right)^{1\cdot28}=\frac{110}{14}=7\cdot86$$

Taking logarithms $\quad 1\cdot28\log\dfrac{117\cdot8+c}{c}=0\cdot895$

$$\log\frac{117\cdot8+c}{c}=0\cdot699$$

$$\frac{117\cdot8+c}{c}=5$$

$$5c=117\cdot8+c$$

$$\underline{c=29\cdot4 \text{ cu. in.}}$$

6. The equation to the compression curve of an oil engine is $pv^{1\cdot33}$ =constant. If the initial pressure is 14 lb. per sq. in. and the initial temperature is 70° C., what must be the volumetric ratio of compression and the final pressure if the final temperature is to be 500° C. approximately ?

$$p_1v_1^{1\cdot33}=p_2v_2^{1\cdot33}$$

also $\quad \dfrac{p_1v_1}{T_1}=\dfrac{p_2v_2}{T_2}$

By division $\quad v_1^{0\cdot33}.T_1=v_2^{0\cdot33}.T_2$

$$\therefore\ \left(\frac{v_1}{v_2}\right)^{0\cdot33}=\frac{T_2}{T_1}$$

$T_1=70+273=343.\quad T_2=500+273=773.\quad \dfrac{T_2}{T_1}=2\cdot25$

$$\frac{v_1}{v_2}=r \text{ (ratio of compression)}$$

$$\therefore\ r^{0\cdot33}=2\cdot25$$

Taking logarithms $\quad 0\cdot33\log r=0\cdot352$

$$\log r=1\cdot067$$

$$\underline{r=11\cdot7 \text{ (about 12)}}$$

EXAMPLES

CHAPTER I

1. A motor car weighing 30 cwt., moving at 45 m.p.h., needs 230,000 ft.-lb. of work done to bring it to rest. How much heat is generated at the brakes during stopping from this speed ?

2. An oil engine uses 15·2 lb. of fuel oil per hour. If each pound of fuel develops 10,600 C.H.U. when completely burned, and 30 per cent. of the heat is converted to useful work, find (*a*) the useful work done per minute, (*b*) the useful horsepower.

3. The weight of steel in a boiler is 12 tons and it contains 9 tons of water. If each pound of coal burned gives up 7000 B.Th.U. to the boiler, what weight of coal must be burned to raise the temperature of the boiler and contents from 65° F. to 370° F. ?

CHAPTER III

1. Sketch a section through a steam engine cylinder showing the ports and position of slide valve when piston is at end of stroke. Details of cover, stuffing boxes, etc., to be shown.

2. Define specific heat, latent heat, total heat of steam. Distinguish between wet, dry saturated, and superheated steam.

CHAPTER IV

1. The vacuum in the condenser of a steam engine is 26 in. of mercury (barometer at 30 in.). What is the highest possible temperature of the air-pump discharge ?

2. What weight of dry saturated steam at 16 lb. per sq, in. abs. must be condensed in a direct contact feed heater in order to raise 1 lb. of the feed water from 60° F. to 212° F. ? What is the maximum possible temperature to which the feed water can be raised in this case ?

3. Of what use is a live steam-feed heater ? How many pounds of steam at 180 lb. per sq. in. abs. (dry saturated) will be required per pound of feed water if the temperature of the feed water is to be raised from 60° to the temperature of the steam ? Assume that a direct contact feed heater is used.

4. An engine of 150 I.H.P. uses 14 lb. of steam per indicated horsepower per hour.

(*a*) If the speed of the engine is 180 r.p.m., find the weight of steam used per stroke.

(*b*) Find the work done per pound of steam.

(*c*) If the dryness of the steam is 0·93 before reaching the separator, and 0·98 after leaving the separator, find the weight of water discharged from the separator per hour.

(*d*) Find the amount of latent heat required to evaporate this quantity of water at atmospheric pressure.

5. Find the heat required to superheat 1 lb. of steam at 160 lb. per sq. in. (by gauge) by 150° F., the mean specific heat of the superheated steam being 0·55.

What percentage of the total heat of the superheated steam from 65° F. is required for superheat ?

6. A steam engine of 160 I.H.P. uses 2880 lb. of steam per hour. The initial pressure of the steam is 155 lb. per sq. in. by gauge (t=368° F., L=854 B.Th.U.) and its dryness is 0·95. If the steam is generated from feed water at 70° F., calculate the indicated thermal efficiency.

7. The following data was obtained from a steam engine trial :

Duration of trial	70 minutes
Total air-pump discharge	1746 lb.
Total revolutions	14,760
Average mean effective pressure	54·6 lb. per sq. in.
Average gauge pressure of steam at stop valve	125 lb. per sq. in.
Dryness of steam at stop valve	0·95
Effective load on brake	420 lb.

Given diameter of cylinder=10 in., stroke=14 in., effective diameter of brake wheel=7·02 ft., find (*a*) the mechanical efficiency, (*b*) the indicated thermal efficiency of the engine assuming that the steam was generated from feed water at 75° F.

8. Find the thermal efficiency of a Diesel engine of 700 B.H.P. using 294 lb. of fuel oil per hour, the calorific value of the oil being 18,500 B.Th.U. per lb.

If the cost of the oil is £3 15*s*. per ton, and the cost of coal of 12,500 B.Th.U. per lb. is £1 10*s*. per ton, how much coal must be used per hour for a steam plant of the same power, if the fuel cost is the same in both cases ?

What would be the overall thermal efficiency of such a steam plant ?

9. In a compound condensing steam engine it is found that 4380 lb. of superheated steam is used per hour when the indicated horsepower is 480. The absolute pressure of the steam at the engine stop valve is 130 lb. per sq. in., and its temperature is 700° F. The temperature of the hot well, which may be taken as the temperature of the boiler feed, is 110° F. Taking the mean specific heat of superheated steam as 0·54, calculate the thermal efficiency of the engine.

CHAPTER V

1. Give a list of the causes of loss of economy in steam engines operated by slide valves, and explain carefully in what way each of these reduces the economy.

2. What is meant by the term 're-evaporation'? Explain clearly why re-evaporation occurs, and the way in which it tends to increase initial condensation.

3. Explain clearly why superheating improves the economy of a reciprocating steam engine, allowing for the fact that extra heat must be given to the steam to superheat it.

4. A cylinder is 30 in. diameter, and 36 in. long. (*a*) What weight of dry steam at 100 lb. per sq. in. abs. will just fill the cylinder?
(*b*) What is the weight of a film of water 0·003 in. thick, deposited on the cylinder ends (regarded as flat) and barrel? (1 cu. ft. of hot water weighs 61·5 lb.)

5. An engine of 750 I.H.P. uses 12 lb. of steam per hour per indicated horsepower at all loads from 750 to 400 I.H.P. The friction horsepower, which may be taken as constant at all loads, is 75. Calculate the steam per brake horsepower-hour at 750 I.H.P. and at 450 I.H.P. What conclusions do you draw from your results?

6. What circumstances must be taken into account in settling the most economical ratio of expansion for a reciprocating steam engine?
The diameter of a steam engine cylinder is 10 in., stroke 15 in., and cut-off is at 0·4 of the stroke. The initial pressure is 140 lb. per sq. in. by gauge and the back pressure is 2 lb. per sq. in. abs. Taking the hypothetical diagram, and neglecting clearance and condensation, calculate the work done per cubic foot of steam used.

7. Explain why it is much more important to have a small clearance volume in an engine operating with a very early cut-off, than in one operating with a late cut-off.

8. It is known that with a certain type of engine, working under given conditions, the work done per pound of steam is practically the same for ratios of expansion of 4, 5, and 6. Which of these ratios would you prefer to adopt in designing a new engine of this type, and why?

9. The diameter of a steam engine cylinder is 12 in., stroke 15 in., revolutions per minute 280. If the cut-off is at 0·5 of the stroke, initial pressure 160 lb. by gauge (dry saturated), back pressure 2 lb. per sq. in. abs., calculate (*a*) the indicated horsepower, assuming a diagram factor of 0·7, (*b*) the steam consumption in pounds per hour, assuming that the clearance is 10 per cent., and that 15 per cent. of the steam entering the cylinder is condensed.

10. Steam at 185 lb. per sq. in. by gauge enters a cylinder and is cut off at 0·4 of the stroke. The back pressure is 15 lb. per sq. in. abs. Assuming hyperbolic expansion and neglecting clearance, compression, and condensation, (*a*) find the mean effective pressure; (*b*) find the work done per pound of steam if 1 lb. of the steam used has a volume of 2·32 cu. ft.; (*c*) assuming that the actual work done per pound of steam is 80 per cent. of this, and that the engine has to supply 5000 lb. of exhaust steam per hour for heating purposes at 15 lb. pressure, find the horsepower developed by a back-pressure engine working under these conditions.

CHAPTER VI

1. Draw a diagram showing the relative positions of piston and crankpin when the crank makes angles of 0°, 30°, 60°, 90°, 120°, 150°, 180°, when the length of the connecting rod is $1\frac{1}{2}$ times the stroke of the piston. Mark also upon the piston path, the piston positions for the same crank angles if the connecting rod were infinitely long.

2. In an engine with a cylinder 24 in. diameter and 3 ft. stroke, the mean pressure of the steam on the piston is 45 lb. per sq. in.; find the mean pressure on the crankpin in the direction of its motion.

3. If cut-off takes place on both sides of a piston when the crank makes an angle of 90° with the dead point, (1) assuming connecting rod infinitely long, (2) assuming connecting rod four times length of crank; find in each case for each side of the piston the fraction of stroke at which cut-off takes place.

4. When steam pressure is acting on a piston, is the whole of it transmitted through the piston-rod to the crosshead? If not, how is the difference employed?

And if the speed of the engine increased while the steam pressure remained about the same, would the force at the crosshead remain the same as before? If not, why not?

CHAPTER VII

1. Define 'outside lap,' 'inside lap,' and 'lead' of a slide valve; make sketches illustrating your answer.

2. The width of a steam port is $1\frac{1}{4}$ in.; the lap of the valve $\frac{9}{16}$ in.; and the lead $\frac{1}{8}$ in. Draw a diagram giving the travel of valve and angular advance of the eccentric.

3. Find the travel of a valve having $\frac{3}{4}$ in. outside lap, and maximum port opening $1\frac{3}{8}$ in.

4. Sketch a slide valve in mid position to the following dimensions: exhaust port 3 in. wide, bars 1 in. wide, steam ports 2 in. wide, outside lap $1\frac{1}{2}$ in. Sketch also the same valve at the beginning of the piston stroke with $\frac{1}{2}$ in. lead.

5. A slide-valve is worked directly from an eccentric. The advance is 30°. When the main crank has moved 20° from the line of centres, show the position of the eccentric crank. The half travel being 3 in., mark off this radius and drop a perpendicular on the line of centres: what have you thus found?

6. A link motion or other gear for a slide valve will reverse an engine, but suppose we do not reverse the engine; suppose we only change from say full to half gear, state clearly what it is that is really effected by the change. Sketch also the probable change in the indicator diagram.

CHAPTER VIII

1. Explain why each of the following tends to improve the economy of a steam engine: (*a*) Separate steam and exhaust ports, (*b*) stage expansion, (*c*) superheated steam.

Show by sketches the method of operating the valves of a drop-valve engine, including the method of varying the point of cut-off.

2. Describe, with the aid of sketches, the construction and working of a uniflow engine, and state in what respects it is more economical than an ordinary drop-valve engine.

If compression starts at 0·1 of the return stroke, the clearance is 3 per cent. of the stroke volume, the back pressure is 1·5 lb. per sq. in. abs., and the equation to the compression curve is $p.v^{1\cdot3}$=constant, calculate the pressure at the end of compression.

3. In what respects are engines with valves operated by trip gears superior to the ordinary slide-valve engine? What considerations should be taken into account in deciding the type of engine to be installed for a particular purpose?

4. An old engine of 860 I.H.P., using 18 lb. of steam per hour per indicated horsepower, is to be replaced by a modern engine of the same power using 12 lb. of steam per hour per indicated horsepower. The boilers generate 8·2 lb. of steam per pound of coal, and the cost of the coal used is 32*s*. per ton.

(*a*) Calculate the saving in a year of 2500 working hours at full power.
(*b*) In what other ways could savings be effected?
(*c*) Taking interest and depreciation at 15 per cent., what capital expenditure on the new plant would be justified?

5. The following data of uniflow engines is given:

	Diameter of cylinder (inches)	Stroke (inches)	Revolutions per minute	I.H.P.	Boiler pressure, lb. per sq. in. gauge
(i)	67	55	120	4000	180
(ii)	35	40	120	1100	200
(iii)	60	72	28	—	180
(iv)	29	33	140	684	175

(*a*) Calculate for (i), (ii), and (iv) the mean effective pressure in each case.

(*b*) Calculate the indicated horsepower for (iii), assuming a mean effective pressure of 35 lb. per sq. in.

(*c*) Assuming a vacuum of 28 in. (30 in. barometer), calculate the maximum load on the piston in tons in each case.

6. Explain carefully why the use of superheated steam improves the steam consumption of (*a*) a reciprocating steam engine, (*b*) a steam turbine. What type of reciprocating steam engine is most suitable for the use of highly superheated steam ? Sketch the admission valve and seating of this type of engine.

CHAPTER X

1. What are the objects of fitting a condenser to a steam engine or turbine ? Jet condensers are largely used in land practice, and surface condensers exclusively in marine practice. Explain why this is so, and give sketches showing the construction of a good condenser of either type.

2. Compare the advantages and disadvantages of jet and surface condensers respectively.

An engine of 1200 I.H.P. uses 18 lb. of steam per hour per indicated horsepower. If the terminal pressure in the low-pressure cylinder is 8 lb. per sq. in. abs. (dryness 0·8) and the rise of temperature of the cooling water is 30° F., find the number of pounds of cooling water required per hour if the condensed steam leaves at 100° F.

3. A steam turbine of 20,000 B.H.P. uses 9 lb. of steam per brake horsepower-hour at full load. The steam enters the condenser at a pressure of 1 lb. per sq. in. abs. ($t=102°$ F., L$=1033$ B.Th.U.), its dryness being 0·85. The rise of temperature of the cooling water is 28° F. and the temperature of the condensed steam at exit from the condenser is 88° F. Calculate the weight of cooling water required per minute.

4. Give sketches showing the construction of some form of condenser suitable for a high vacuum.

The exhaust steam from an engine of 5000 H.P., using 12 lb. of steam per hour per horsepower, enters the condenser at 1·5 lb. per sq. in. abs. ($t=115°$ F., L$=1026$ B.Th.U.), its dryness being 0·85. The exit temperature of the condensed steam is 95° F. If 60 lb. of cooling water is circulated per pound of steam, calculate the rise of temperature of the cooling water and the weight to be circulated per minute.

5. Describe, with the aid of sketches, the construction and working of an air pump suitable for the production of a high vacuum.

CHAPTER XI

1. Give reasons why the loaded Porter governor has been displaced in modern engines by governors of the spring loaded type.

In a Porter governor the weight of each ball is 7 lb. and the load on the sleeve is 49 lb. (*a*) Find the 'height' of the governor for a speed of 240 r.p.m., neglecting friction. (*b*) If the equivalent friction at the sleeve is 2 lb., find the two speeds between which the governor will not act, the height being as found in (*a*).

2. The weight of each ball of a Hartnell type governor is 5 lb., and its radius of action at a speed of 300 r.p.m. is 5 in. What is the centrifugal force on each ball at this speed? If the two arms of the bell-crank lever are 6 in. and $3\frac{1}{2}$ in. long, what total load must the spring exert on the sleeve at this speed?

3. Distinguish clearly between the action of a flywheel and that of a governor.

A flywheel stores 240,000 ft.-lb. of energy at a speed of 275 r.p.m. How much work can it do in changing its speed to 270 r.p.m.?

CHAPTER XII

1. A locomotive exerts 2400 H.P. in drawing a train at 50 m.p.h. If it burns 1·8 lb. of coal per hour per horsepower, what weight of coal will be burned in a journey of 350 miles at the above speed?

CHAPTER XIII

1. Give sketches showing the construction of a modern indicator. Show clearly how the motion of the indicator piston is magnified and a straight-line motion of the pencil obtained.

2. Sketch a suitable reducing gear for an indicator and state what precautions must be taken in fitting it up. State the possible faults in the indicating mechanism, and give typical diagrams showing the effects of these faults.

3. What are the objects of taking 'light spring' diagrams from an internal combustion engine? What information may be obtained from such diagrams?

CHAPTER XIV

1. State the disadvantages of very early cut-off in the cylinder of a single-expansion engine, and explain clearly in what way these disadvantages are obviated by the use of successive expansion. What other advantages are obtained?

2. Give a sketch showing the arrangement of the cylinders of a set of triple-expansion engines, showing the path of the steam from the boilers to the condenser.

What modification of the steam piping is necessary if two low-pressure cylinders of the same size are used ?

3. A compound engine is to develop 900 I.H.P. at 120 r.p.m. The initial pressure is 200 lb. per sq. in. by gauge, the back pressure in the low-pressure cylinder is 4 lb. per sq. in. abs., the total ratio of expansion is to be 8, and the diagram factor is 0·8. If the ratio of cylinder volumes is 1 : 5, calculate (*a*) the diameters of the cylinders, (*b*) the point of cut-off in the high-pressure cylinder. The stroke is to be 24 in.

4. A compound engine is to develop 500 I.H.P. at 160 r.p.m. The initial pressure is 200 lb. per sq. in. by gauge, back pressure in low-pressure cylinder 2 lb. per sq. in. abs., total ratio of expansion 12, ratio of cylinder volumes 1 : 7. Taking a diagram factor of 0·75 and stroke=0·7 of low-pressure cylinder diameter, calculate the necessary diameters of the cylinders and the point of cut-off in the high-pressure cylinder.

5. A set of triple-expansion engines is to develop 2200 I.H.P. at 120 r.p.m., the total ratio of expansion being 12. The boiler pressure is 185 lb. per sq. in. by gauge, the back pressure in the low-pressure cylinder is 3 lb. per sq. in. abs., and the stroke is 3 ft. Ratio of cylinder volumes 1 : 3·4 : 7·2.

Taking a diagram factor of 0·7, calculate

(*a*) the diameter of the low-pressure cylinder ;
(*b*) the point of cut-off in the high-pressure cylinder ;
(*c*) the diameters of the high-pressure and medium-pressure cylinders.

6. A compound engine is to develop 750 I.H.P. at 120 r.p.m. The initial pressure is 200 lb. per sq. in. by gauge and the back pressure in the low-pressure cylinder is 2 lb. per sq. in. abs. The total ratio of expansion is 10 and the stroke is to be two-thirds (approximately) of the diameter of the low-pressure cylinder. Ratio of cylinder volumes=1 : 6. Calculate the necessary diameters of the high-pressure and low-pressure cylinders, also the stroke. Assume a diagram factor of 0·75.

7. Explain in what respects the modern cross-compound drop-valve engine, using superheated steam, is superior to an engine of similar power worked by slide valves, and using saturated steam. What is the effect of using an earlier cut-off in the low-pressure cylinder only, (*a*) on the total horsepower, (*b*) on the distribution of power between the cylinders ? Illustrate your answer by means of a diagram.

CHAPTER XV

1. A boiler generates 7·8 lb. of steam per pound of coal. The pressure of the steam is 315 lb. per sq. in. abs., superheated 250° F.

(saturation temperature 422° F., L=815 B.Th.U.), and the temperature of the feed is 195° F. If the calorific value of the fuel is 11,600 B.Th.U. per lb., calculate the efficiency of the boiler. (Mean specific heat of superheated steam=0·57.)

2. Give a list of the causes of loss of efficiency in a boiler, and state the methods which are adopted in practice to reduce these losses.

A boiler generates steam at 365 lb. per sq. in. abs. (ts=436° F., L=802 B.Th.U.), superheated 220° F., from feed water at 90° F. If the efficiency of the boiler is 82 per cent. and the calorific value of the fuel is 12,600 B.Th.U. per lb., calculate the weight of steam generated per pound of fuel. (Mean specific heat of superheated steam=0·57.)

3. A boiler supplies 12,000 lb. of steam per hour at 250 lb. per sq. in. by gauge, superheated 200° F. (c=0·57), generated from feed water at 75° F. If its efficiency is 85 per cent., and the calorific value of the fuel is 12,250 B.Th.U. per lb., calculate the weight of fuel consumed per hour.

4. State what devices are used in practice to improve the efficiency of boilers, and give sketches showing the general construction of one of these devices.

A boiler generates steam at 250 lb. per sq. in. by gauge, superheated 180° F., from feed water at 85° F. If the efficiency of the boiler is 78 per cent., calculate the weight of water evaporated per pound of fuel, the calorific value being 12,400 B.Th.U. per lb. (Mean specific heat of superheated steam=0·56.)

5. A certain boiler generates 8·4 lb. of steam per pound of fuel from feed water at 70° F. The steam is generated at 185 lb. per sq. in. by gauge (t=382° F., L=849 B.Th.U.), and is superheated 160° F. (c=0·57). If the calorific value of the fuel is 12,600 B.Th.U. per lb., what is the efficiency of the boiler ?

If the air enters the boiler at 80° F., and the flue gases leave at 350° F., and 20 lb. of air is supplied per pound of fuel burned, what percentage of the heat of combustion is lost in the flue gases ? (Specific heat=0·24.)

6. Give sketches showing the construction of some form of superheater, and show by a diagram its approximate position in a boiler of any type.

Steam at 300 lb. per sq. in. abs. is generated from feed water at 220° F. Find the percentage increase of heat required to superheat the steam by 300° F., taking the mean specific heat of superheated steam to be 0·58.

7. State in what ways economy is effected by the use of superheated steam in (a) a reciprocating steam engine, (b) a steam turbine.

A boiler is to generate 12,000 lb. of steam per hour at a pressure of 360 lb. per sq. in. abs., superheated 300° F. (ts=435° F., L=804 B.Th.U., specific heat=0·57), from feed water at 240° F. If the efficiency of the boiler is 82 per cent., calculate the weight of coal burned per hour. (Calorific value=11,800 B.Th.U. per lb.)

8. Show by means of a sketch the main features of construction of a boiler fitted with economiser and air heater.

The following observations are made during a boiler trial : Calorific value of fuel, 11,800 B.Th.U. per lb. ; 18 lb. of air supplied per pound of fuel ; temperature of gas at entrance to and exit from economiser, 700° and 450° F. respectively ; temperature of gases at exit from air heater, 280° F. Taking the mean specific heat of the gases as 0·26, calculate (*a*) the percentage of total heat of combustion saved by the economiser, (*b*) the percentage saved by the air heater.

9. A boiler is fired with coal of calorific value 12,000 B.Th.U. per lb. The efficiency of the boiler alone is 65 per cent. and 24 lb. of air is supplied per pound of fuel. If the losses due to incomplete combustion, radiation and heat in ashes, etc., amount to 8 per cent. and the air enters the boiler at 70° F., calculate the approximate temperature of the flue gases at exit from the boiler. (Specific heat=0·24.)

State in what ways the efficiency of the boiler plant may be increased.

10. An engine of 1500 horsepower uses 15 lb. of steam per hour per horsepower. If the engine works on full load for 10 hours a day 300 days per year, and 1 lb. of fuel generates 8 lb. of steam, (*a*) find the annual cost of fuel at 25*s*. per ton. (*b*) If the steam consumption is reduced to 12 lb. per hour per horsepower, find the annual saving in fuel cost. (*c*) If the boiler is now made to evaporate 8·5 lb. of water per pound of fuel, the cost per ton remaining the same, find the additional annual saving.

11. Compare the efficiencies of two boilers A and B given the following particulars :

A. 8·1 lb. of steam per pound of fuel at 200 lb. per sq. in. (gauge), dryness 0·92, from feed water at 70° F., calorific value of fuel 12,500 B.Th.U. per lb.

B. 8·6 lb. of steam per pound of fuel at 160 lb. per sq. in. (gauge), superheated 160° F. (c=0·55), from feed water at 230° F. Calorific value of fuel 13,000 B.Th.U. per lb.

Under what circumstances is the comparison (*a*) fair, (*b*) unfair ?

CHAPTER XVI

1. A sample of fuel is to be tested for calorific value. Give sketches of the apparatus used, describe the method of conducting the experiment, and show how the observations taken during the experiment are used in calculating the final result.

2. A sample of coal contains 85 per cent. of carbon and 4·5 per cent. of available hydrogen, and is burned with 40 per cent. excess air. Calculate from first principles the weights of CO_2, oxygen, nitrogen, and steam in the products of combustion of 1 lb. of this fuel. What is the percentage of CO_2 by weight in these products ?

3. The following observations were made during a test of the calorific value of a sample of coal :

Weight of crucible	=1·732 grams
Weight of crucible and coal	=2·452 ,,
Weight of water in calorimeter	=1400 ,,
Water equivalent of calorimeter	= 220 ,,
Rise of temperature after combustion of coal	=3·18° C.

The radiation loss is negligible.

Calculate the calorific value of the coal in C.H.U. and B.Th.U. per pound. (1 lb.=454 grams.)

Make a sketch of some form of apparatus in which an experiment of this nature would be conducted, and explain briefly how it would be used.

4. 1 lb. of fuel contains 0·82 lb. of carbon and 0·06 lb. of hydrogen. Calculate from first principles the minimum weight of air required for combustion. (Air contains 23 per cent. by weight of oxygen.) If the fuel is burned with 60 per cent. excess of air, calculate the percentage by weight of CO_2 in the products of combustion. What other constituents are present in these products ?

5. The cylinder of a Diesel engine is 28 in. diameter, and the stroke is 34 in. The weight of 1 cu. ft. of air at suction pressure and temperature is 0·074 lb. If the fuel oil contains 86 per cent. of carbon and 12 per cent. of hydrogen, find from first principles the maximum possible weight of fuel oil that can be burned per stroke. State why only about 60 per cent. of this is practicable if combustion is to be complete.

6. A boiler burning 1200 lb. of coal per hour (calorific value 12,200 B.Th.U. per lb.) has an efficiency of 64 per cent. without an economiser. An economiser is fitted which utilises one-half of the waste heat. If 9800 lb. of steam is generated per hour and the feed water enters the economiser at 65° F., to what temperature will the feed water be raised in the economiser ?

7. Write down the chemical equations which show the combustion of carbon and hydrogen respectively. Using these equations, find the minimum weight of oxygen required to burn 1 lb. of a fuel containing 0·85 lb. of carbon and 0·06 lb. of hydrogen. If this fuel is burned with 60 per cent. of excess air, calculate the percentage of CO_2 by weight in the products of combustion. (Air contains 23 per cent. by weight of oxygen.)

8. One pound of fuel oil, containing 0·86 lb. of carbon and 0·12 lb. of hydrogen, is burned completely with 24 lb. of air. Calculate the weights of CO_2, steam, oxygen, and nitrogen in the products of combustion. (1 lb. of air contains 0·23 lb. of oxygen.)

9. A boiler burns 1200 lb. of coal per hour, and 18 lb. of air is supplied per lb. of fuel burned. If the air enters at 65° F., and the flue gases leave the boiler at 650° F., how much heat is wasted per hour in the flue gases ? The specific heat of these gases may be taken as 0·24. If half of this waste heat is utilised in heating feed water, to what temperature could 9600 lb. of water be raised from 55° F.

10. Two samples of coal, A and B, are submitted for decision regarding the use of one of these with a large boiler plant. Their properties and prices are as follows :

Sample A. Calorific value 13,500 B.Th.U. per lb., ash 5 per cent., price 35*s*. per ton.

Sample B. Calorific value 11,200 B.Th.U. per lb., ash 12 per cent., price 28*s*. 6*d*. per ton.

State all the considerations you would take into account in deciding which tender to accept.

11. State the causes of production of smoke by a boiler. In what ways may smoke be reduced or eliminated ? What other product of incomplete combustion may be present in the flue gases, and in what ways is its presence undesirable ?

CHAPTER XVIII

1. State the principles underlying the working of a steam turbine. Distinguish between 'impulse' and 'reaction' turbines and state the advantages of using a large number of stages in each.

Why is a very high vacuum of much greater importance for a turbine than for a reciprocating engine ?

2. The velocity of issue from the nozzles of one of the stages of an impulse steam turbine is 1200 ft. per second, and the inclination of the nozzles is 20°. If the speed of the blading is 500 ft. per second, (*a*) find the necessary angles of the blading at entrance and exit, (*b*) neglecting losses, find the absolute velocity of the steam at exit from the blading, (*c*) find the work done per pound of steam.

Repeat the calculations for blading velocities of 600 ft. per second and 700 ft. per second.

3. Give reasons why, with similar initial steam conditions, the steam turbine is so much more economical than the reciprocating steam engine. Sketch the general arrangement of a turbine of either the impulse or the reaction type.

4. At a certain stage of an impulse steam turbine, steam issues from the nozzles at 1250 ft. per second, the inclination of the nozzles being

20°. The mean diameter of the blading is 5 ft. 9 in. and the speed of revolution is 1500 per minute. Neglecting losses, find (*a*) the correct angles of entry and exit for the blading, (*b*) the magnitude and direction of the velocity of the steam at exit from the blading.

5. At a certain stage of an impulse steam turbine the mean speed of the blading is 750 ft. per second. The inclination of the nozzles is 20° and the steam issues at 1800 ft. per second. The quantity flowing is 30 lb. per second. Find (*a*) the necessary angles at entry and exit of the blading, (*b*) the magnitude and direction of the exit velocity from the wheel, (*c*) the work done per second, neglecting losses.

CHAPTER XIX

1. 0·2 cu. ft. of coal gas is mixed with 0·7 cu. ft. of air at 15 lb. per sq. in. abs., the mixture being at 200° F. The mixture is then compressed to a volume of 0·2 cu. ft., the pressure then being 95 lb. per sq. in. abs. What will then be the temperature ? Explosion now takes place at constant volume, and the pressure rises to 340 lb. per sq. in. abs. What is now the final temperature ?

2. 1 cu. ft. of air at 18° C. is to be expanded until the temperature falls to $-20°$ C. If the expansion follows the law $p.v^{1\cdot 3}$=constant, to what volume must it be expanded ? Show that the result is independent of the initial pressure.

3. 0·038 lb. of oil of calorific value 19,000 B.Th.U. per lb. is burned in a cylinder with 1 lb. of air initially at 775° F. If 22 per cent. of the heat is dissipated by radiation and conduction, to what temperature will the air be raised ? (Mean specific heat=0·28.)

If the initial pressure of the air is 275 lb. per sq. in., what will be the final pressure ? (You may assume that the volume remains constant during the combustion.)

4. The volumetric compression ratio of an oil engine is 11·2 and the equation to the compression curve is $p.v^{1\cdot 32}$=constant. If the pressure and temperature at the beginning of compression are 13·5 lb. per sq. in. and 80° C. respectively, calculate the pressure and temperature at the end of compression.

CHAPTER XX

1. The following results were obtained from the test of an oil engine :

Duration of test	=55 minutes
Weight of oil used	=5·79 lb.
Average I.H.P.	=18·1
Average B.H.P.	=14·24

Calorific value of fuel=19,100 B.Th.U. per lb.

Calculate (*a*) the indicated thermal efficiency, (*b*) the brake thermal efficiency.

2. The following data was obtained from the trial of a single-cylinder gas engine :

Diameter of cylinder, 6·7 in. ; stroke, 15 in. ; brake radius, 27 in.

Average mean effective pressure	=96·4 lb. per sq. in.
Average revolutions per minute	=194
Average explosions per minute	=54·4
Net brake load	=50 lb.
Gas used per minute	=2·6 cu. ft.
Weight of cooling water per minute	=4·78 lb.
Rise of temperature	=55° C.
Calorific value of gas	=430 B.Th.U. per cu. ft.

Calculate (i) gas per hour per indicated horsepower, (ii) gas per hour per brake horsepower, (iii) indicated thermal efficiency, (iv) percentage of heat to cooling water.

3. The indicated thermal efficiency of an internal combustion engine of 2000 I.H.P. is 0·43. If 60 per cent. of the waste heat can be utilised for the production of steam, find the number of pounds of dry saturated steam per hour which can be generated at 175 lb. per sq. in. absolute from feed water at 70° F.

If 1 I.H.P. can be generated from 20 lb. per hour of this steam, find the possible increase in horsepower and state the percentage increase of power.

4. A marine steam engine of 2600 B.H.P. uses 1·6 lb. of coal per hour per brake horsepower. An oil engine of the same power uses 0·38 lb. of fuel per hour per brake horsepower. (*a*) Find the weight of fuel in tons saved by the oil engine as compared with the steam engine in a voyage of 240 hours' duration at full power. (*b*) If the coal costs 18*s*. per ton and fuel oil costs £3 10*s*. per ton, find the saving in fuel cost for the voyage. What other advantages accrue from the use of oil engines for marine propulsion ?

5. The diameter of the cylinder of a four-cylinder single-acting Diesel engine is 30 in. and the stroke is 36 in. The indicated mean effective pressure is 95 lb. per sq. in. and the speed is 120 r.p.m. (*a*) Calculate the indicated horsepower. (*b*) If the indicated thermal efficiency is 0·42 and the calorific value of the fuel is 19,200 B.Th.U. per lb., calculate the weight of fuel injected into each cylinder per working stroke.

6. A four-cylinder petrol engine is to develop 32 B.H.P. at 2500 r.p.m. The stroke is to be 1·3 times the bore, the mean effective pressure on each piston is 125 lb. per sq. in. and the mechanical efficiency is 0·87. Calculate the necessary bore and stroke of each cylinder.

7. State the advantages of internal combustion engines for marine propulsion.

A six-cylinder Diesel engine is to develop 2500 B.H.P. at 120 r.p.m. with an indicated mean effective pressure of 110 lb. per sq. in. and a mechanical efficiency of 0·8. The stroke is to be 1·4 times the bore. (*a*) Calculate the necessary bore and stroke in inches. (*b*) If the indicated thermal efficiency is 0·42 and the calorific value of the fuel is 19,200 B.Th.U. per lb., calculate the weight of the fuel used per hour per indicated horsepower.

8. A gas engine when developing 32 I.H.P. uses 581 cu. ft. of gas per hour, the calorific value of the gas being 460 B.Th.U. per cu. ft. The initial temperature of the jacket water is 67° F., and the temperature on leaving the jacket is 162° F. The amount of water flowing through the jacket is 16·4 lb. per minute. Find (*a*) the percentage of the heat supplied which is converted to work, (*b*) the percentage of heat absorbed by the jacket.

9. A single-cylinder high-compression oil engine is to develop 45 B.H.P. at 260 r.p.m. with an indicated mean effective pressure of 95 lb. per sq. in., the mechanical efficiency being 0·8. If the stroke is 16 in., calculate the necessary diameter of the cylinder.

If the brake thermal efficiency of the engine is 34 per cent. and the calorific value of the fuel used is 19,200 B.Th.U. per lb., estimate the weight of oil used per hour at full load.

10. State the conditions that must be satisfied for complete combustion of the fuel in an internal combustion engine using heavy fuel, and explain how these may be satisfied in practice.

The pressure and temperature at the beginning of the compression stroke of a Diesel engine are 14 lb. per sq. in. and 180° F. respectively. The equation to the compression curve is $p.v^{1 \cdot 33}$=constant. Calculate the necessary compression ratio if the temperature at the end of compression is to be 1050° F.

11. The mechanical efficiency of a single-cylinder gas engine, running at a constant speed for all loads, is 80 per cent. when developing 28 B.H.P The friction horsepower and gas consumption per hour per indicated horsepower are the same for all loads If the gas consumption at 28 B.H.P. is 22 cu. ft. per hour per brake horsepower, what is the fuel consumption per brake horsepower-hour when developing 12 B.H.P. ?

If the calorific value of the gas is 500 B.Th.U. per cu. ft., what is the brake thermal efficiency when developing 12 B.H.P. ?

12. A four-cylinder petrol engine is to develop 45 B.H.P. at 2000 r.p.m. The ratio of stroke to bore is 1·4, the mean effective pressure is 125 lb. per sq. in., and the mechanical efficiency is 0·8. Calculate the necessary bore and stroke.

13. A four-stroke single-acting gas engine using coal gas is to develop 30 B.H.P. at 220 r.p.m. with an indicated mean effective

pressure of 95 lb. per sq. in., the mechanical efficiency being 0·8. If the stroke is 18 in., calculate the necessary diameter of the cylinder.

If the thermal efficiency of this engine, based on the indicated horse-power, is 32 per cent., how much gas of calorific value 490 B.Th.U. per cu. ft. will the engine require per hour when carrying the above load?

14. A six-cylinder petrol engine is to develop 45 B.H.P. at 2200 r.p.m. with an average piston speed of 1500 ft. per minute. The mechanical efficiency is 82 per cent. and the indicated mean effective pressure is 110 lb. per sq. in. Calculate the necessary bore and stroke.

ANSWERS

I

1. 295 B.Th.U. (164 C.H.U.). **2.** 1,190,000 ft.-lb. ; 36·1 h.p. **3.** 1019 lb.

IV

1. 126° F. **2.** 0·157 lb. ; 216° F. **3.** 0·366. **4.** (*a*) 0·097 lb. ; (*b*) 141,500 ft.-lb. ; (*c*) 107 lb. ; (*d*) 103,600 B.Th.U. (57,500 C.H.U.). **5.** 82·5 B.Th.U. ; 6·6%. **6.** 12·7%. **7.** (*a*) 0·926 ; (*b*) 0·099. **8.** 0·328 ; 735 lb. ; 0·194. **9.** 0·213.

V

4. 3·31 lb. ; 0·51 lb. **5.** 13·3 lb. ; 14·4 lb. **6.** 42,000 ft.-lb. **9.** (*a*) 245 ; (*b*) 14,800 lb. **10.** 123·2 lb. sq. in. ; 103,300 ft.-lb. ; 209.

VI

2. 12,960 lb.

VII

3. $4\frac{1}{4}$ in.

VIII

4. (*a*) £1124 ; (*b*) £7500. **5.** (*a*) 34, 47·2, 44·5 lb. sq. in. ; (*b*) 1008 ; (*c*) 305, 92, 246, 56.

X

2. 628,000. **3.** 95,800. **4.** 14·9° F. ; 60,000 lb.

XI

1. 4·82 in. ; 244 and 236 r.p.m. **2.** 64·3 lb. ; 220 lb. **3.** 9000 ft.-lb.

XII

1. 13·5 tons.

XIV

3. (*a*) 15·6 in., 35·5 in. ; (*b*) $\frac{5}{8}$. **4.** 11 in., $29\frac{1}{4}$ in. ; 0·58. **5.** (*a*) 57·9 in. ; (*b*) 0·6 ; (*c*) $21\frac{5}{8}$ in., 40 in. **6.** $14\frac{1}{2}$ in., 36 in. ; 24 in.

XV

1. 79·7%. **2.** 8·15 lb. **3.** 1460 lb. **4.** 7·73 lb. **5.** 83·5 %; 10·8%. **6.** 17·1%. **7.** 1454 lb. **8.** (*a*) 10·5% ; (*b*) 7·1%. **9.** 610° F.

10. (*a*) £942 ; (*b*) £220 ; (*c*) £42 10*s*. 0*d*. **11.** Boiler A, 71% ; boiler B, 72%.

XVI

2. CO_2, 3·13 lb. ; O_2, 1·05 lb. ; N_2, 12·28 lb. ; steam, 0·405 lb. ; CO_2, 18·5%. **3.** 7160 C.H.U. ; 12,880, B.Th.U. **4.** 16·2% **5.** 0·0633 lb. **6.** 334° F. **7.** 2·75 lb. ; 16·4%. **8.** 3·15 lb. ; 1·08 lb. ; 2·28 lb. ; 18·48 lb. **9.** 3,200,000 B.Th.U. ; 221·5° F.

XVIII

4. 30·6° ; 507 f. sec. at 58°. **5.** (*a*) 33.5° ; (*b*) 642 f. sec. at 74·2° ; 130,000 ft.-lb.

XIX

1. 469° F. ; 2860° F. **2.** 1·583 cu. ft. **3.** 2785° F. ; 723 lb. sq. in. **4.** 328 lb. sq. in. ; 492° C.

XX

1. 38·3% ; 30%. **2.** 22·3 cu. ft. ; 37·6 cu. ft. ; 26·5% ; 42·4%. **3.** 3480 lb. **4.** (*a*) 341 tons ; (*b*) £29. **5.** 1460 ; 0·032 lb. **6.** 2·84 in., 3·69 in. **7.** (*a*) 28¾ in., 40 in. (*b*) 0·315 lb. **8.** (*a*) 30·5% ; (*b*) 35·7%. **9.** 12 in. ; 17·47 lb. **10.** 13·5. **11.** 27·9 cu. ft. ; 18·3%. **12.** 3·43 in., 4·8 in. **13.** 10·1 in. ; 607 cu. ft. **14.** 3·05 in. ; 4·1 in.

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